PRODUCT ENGINEERING

OCTOBER, 1948

SLEEVE BEARING DESIGN

THIS ISSUE — Current Design Practice in Sleeve Bearings + Analysis of Press Performance + Ni-Span Alloys + Contour Forming + Die Casting Fundamentals

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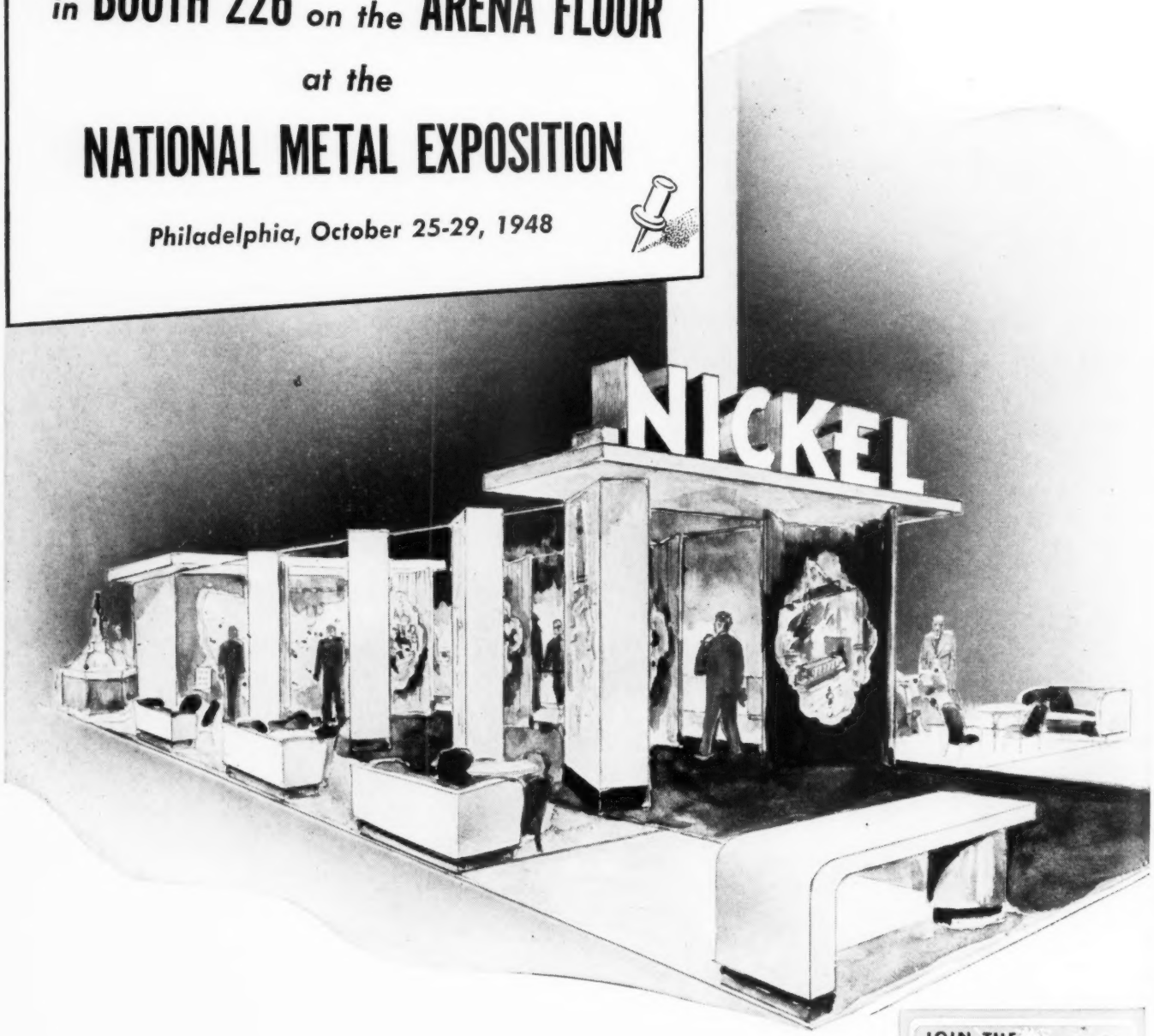

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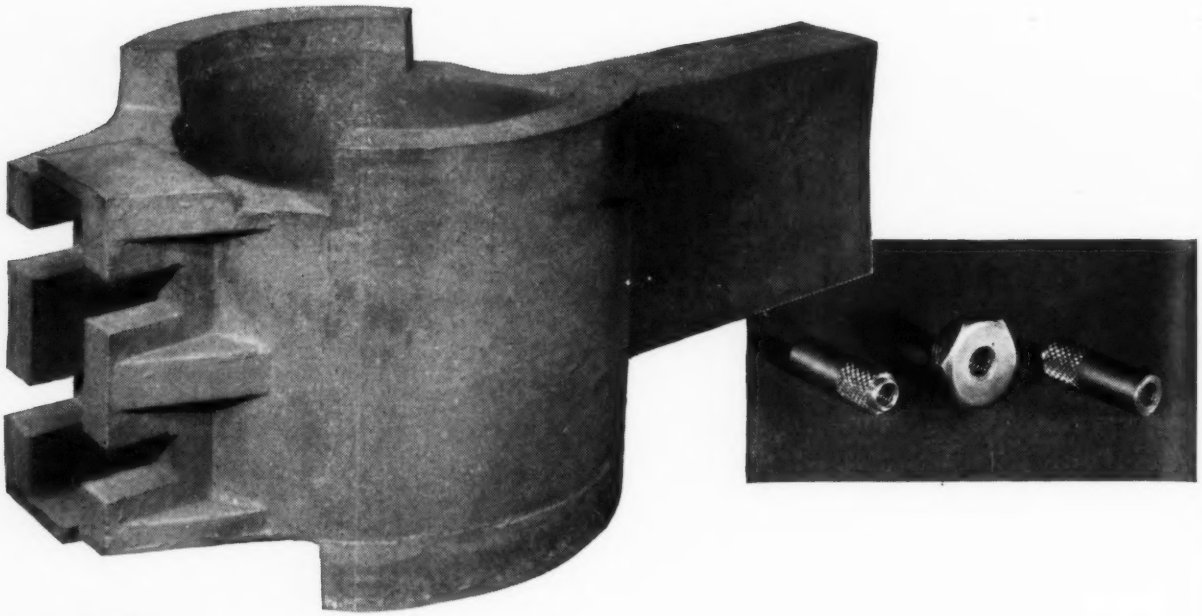

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Design Expense or Creative Investment

NOT MANY DECADES AGO the fund of design knowledge was relatively limited, few materials were available and production machinery and processes were largely confined to metal cutting, grinding and forming. With these limitations the creative possibilities in engineering design were greatly restricted. The output of the design department was measured in blueprints and their cost per square inch was the gage for the efficiency of operation of the engineering department. Design expense was usually considered a necessary evil.

Too many engineering departments are still being operated as a 1900 Model with reference to engineering costs. Management incessantly hammers away at reducing those costs to a minimum. The engineers are impressed with the fact that they must produce lines to hold their jobs and therefore function primarily as a machine for the production of blueprints. Little time is allowed for creative thinking, with the result that the designs run in deeply grooved channels.

Mindful of the tremendous fund of scientific and technical knowledge now available, together with an almost boundless variety of materials and manufacturing techniques, most progressive manufacturers now take an entirely different view of the cost of engineering design and product developments. They recognize that any given product can be produced in an almost endless number of different designs, and each design can be manufactured in many ways. From experience they know that doors are wide open for the inventing and patenting of products in every field of metal manufacturing.

In addition to the profits from the sale of patented products there is always the possibility of royalties. A modest profit on each unit produced by all licensees often totals much more than could be obtained by retaining sole manufacture. There are quite a few companies who derive a large portion of their profits from the licensing of patented products or processes.

Some manufacturers excuse their stagnation in design development with the statement that they are too small to engage in research and product development. Admittedly, many companies are too small to organize and operate a full fledged development department. However, in view of the facilities of commercial research laboratories and university institutes of research, there is no reason for any company engaged in a profitable enterprise not to have a definite product development program. If they are so poor that they cannot even take advantage of outside research facilities available to them, they will either remain small or go out of business.

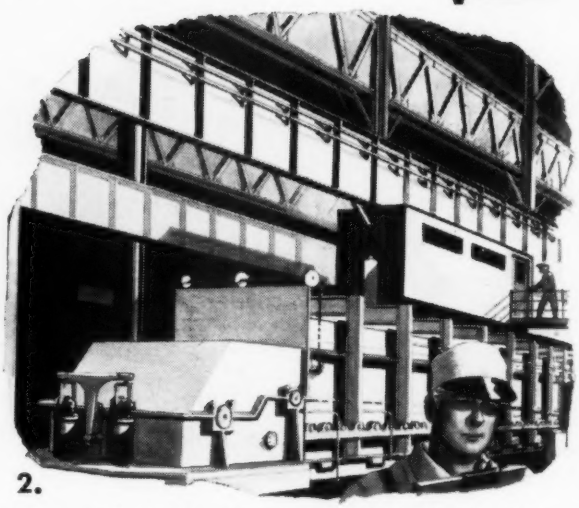
No manufacturer can afford to look upon engineering development costs as an evil overhead or a non-profitable operating expense. There is only one avenue of progress for any manufacturer and that is through the creation of improved products manufacturable at lower cost. The cost of operation of genuine product development program is not a design expense but a creative investment.

G. F. Jordenholt

Hot ideas mean cold cash for you!

1.

"Shut 'er down!" How steel makers hated those words. For every shut-down caused by failure of the steel feeding mechanism made ton-costs mount. High temperatures, giant loads and speed—all major factors in feeding steel to heat-treating furnaces—had bad effects on power transmission equipment. Sit and wait? Not steel makers! They swung into action to beat the heat. They consulted . . .



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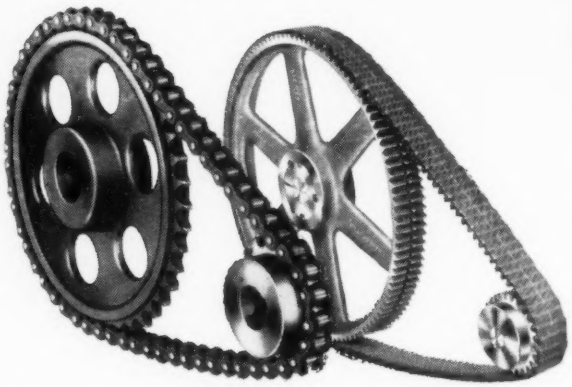

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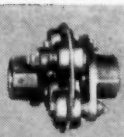
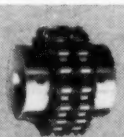
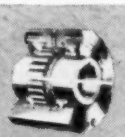


Morflex Couplings



Roller Chain Couplings



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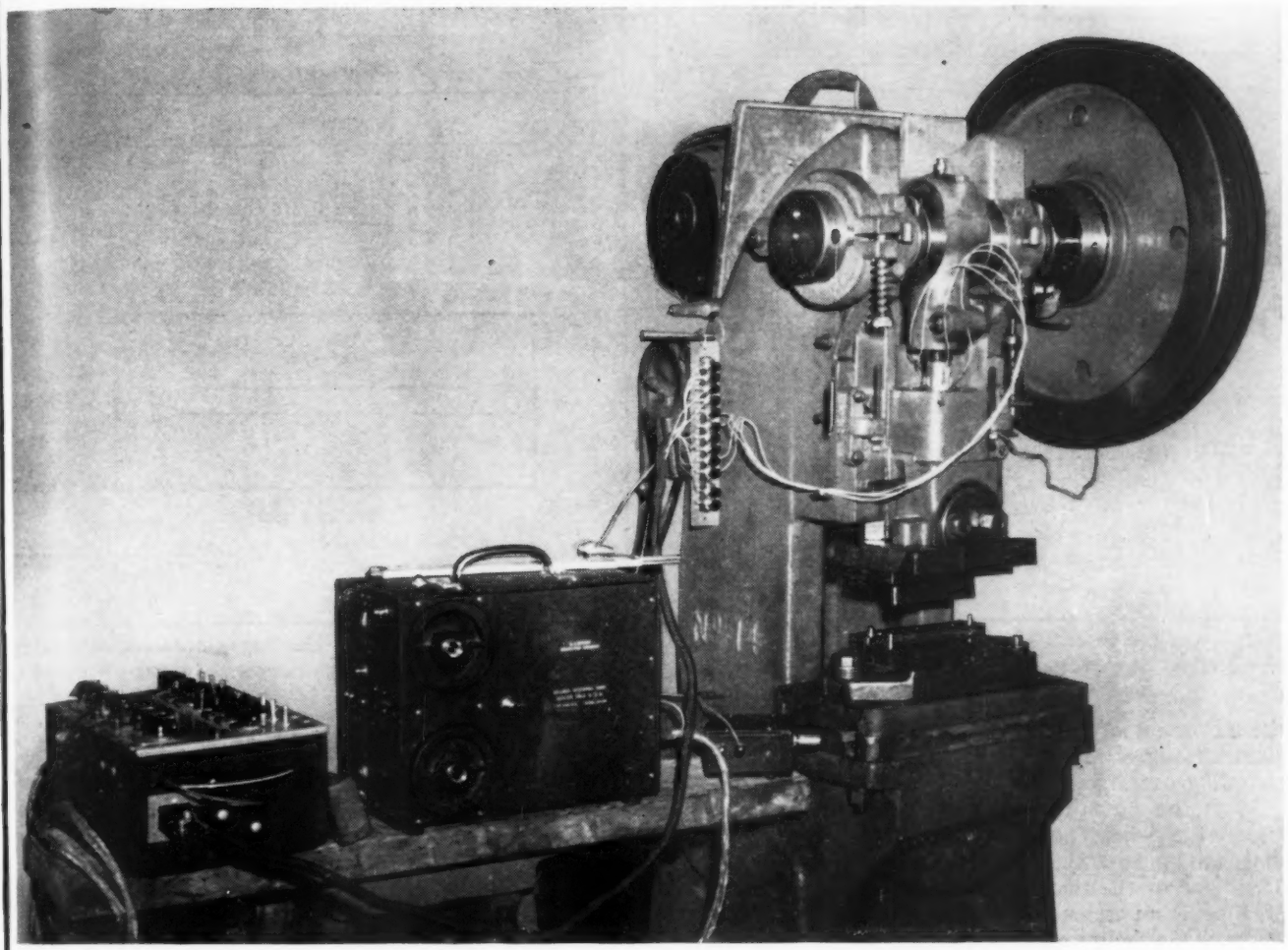


TEST SET UP to study the performance of 14 ton punch press. Electric strain gages are cemented to the ball screw connecting rod.

Analysis of Press Performance

Investigation of performance characteristics of a punch press under service conditions. Study of test data reveals that press performance could be doubled by reducing the overall coefficient of friction in the bearings without changing design of other parts.

GIVEN A. BREWER
Consulting Engineer

PUNCH PRESS DESIGN has long suffered from a lack of accurate data concerning coefficients of friction, flywheel inertia, speed regulation and power expended in useful work. Realizing the need for this information the Diamond Machine Tool Company decided to conduct tests that would yield exact working data. As a result, theoretical formulas were developed that are in good agreement with measured press performance.

For the 14 ton press tested, calculations indicate that in the frame and throw bearings the average coefficient of friction was 0.083. In one test run, where a punch load of 13.1 tons was developed with a flywheel slowdown of 64 percent, it was determined that 53.7 percent of the energy given up by the flywheel and engine was lost in friction.

Further objects of the investigation were the estimation of press performance under conditions of reduced bearing friction and increased flywheel inertia, and the effect of initial flywheel speed and engine horsepower on press characteristics.

For the test, a Diamond No. 14 press was set up as shown above. Two electric strain gages were cemented front and back to the ball screw connecting rod. The gages were connected to a Heiland recording oscillograph so that strains in the ball screw, at the point of gage attachment, would be observed as trace deflections on the oscillograph tape. Compressive strains caused a downward trace on the tape.

From the tape trace, it was possible to measure the front and back strains on the ball screw shank during the working stroke. The average strain multiplied by Young's modulus and the cross-sectional area of the shank, at point of gage attachment, equals the punch load. The upper edge of the tape was marked off in units of time, therefore, the punch load at any given instant could be readily determined.

A light wire belt wrapped around the left hub of the press crankshaft was connected to a pulley on potentiometer. The potentiometer pulley could make 10 revolutions before hitting a stop. Fixed resistors were connected to the potentiometer so that

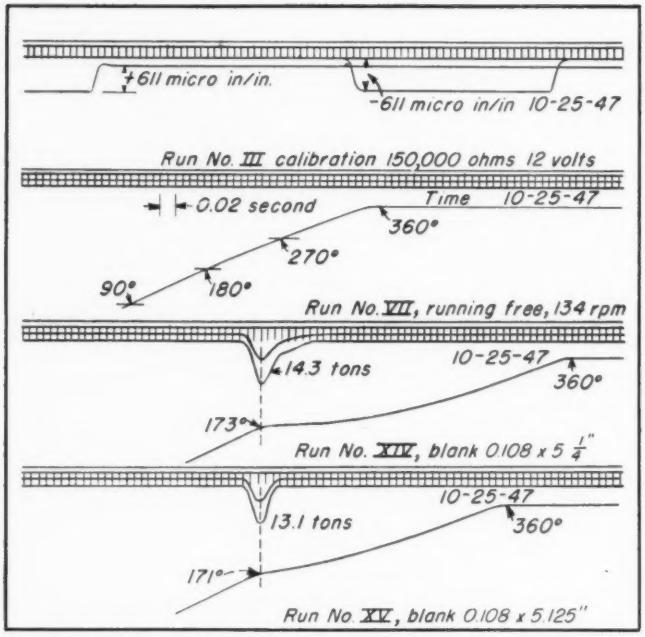


Fig. 1—Oscillograph tape traces for four press runs. Compressive strains are indicated by a downward trace on the tape.

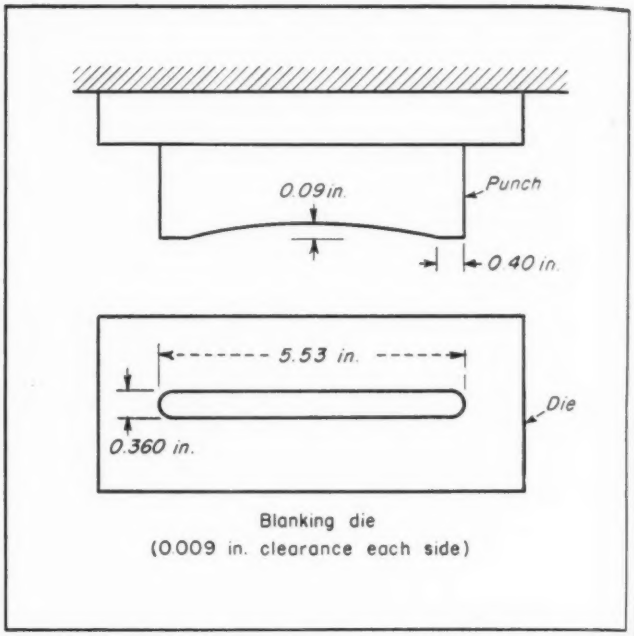


Fig. 2—Dimensions of small blanking die used for blanking hot rolled sheets of different thicknesses in series of test runs.

the movement of the pulley caused a simultaneous trace movement in the recording oscillograph.

With the press running free, the trace of flywheel position advanced diagonally across the chart, Fig. 1. The slope of the trace at any instant of time is an index of the flywheel velocity. On the trace Fig. 1, the position of the flywheel in degrees from top center is marked.

A small blanking die, Fig. 2, was attached to the press and a series of runs were made in which hot rolled sheets of different thicknesses were blanked. The hardness of the hot rolled sheets was 47½ Rockwell B scale. Where R_b is the Rockwell B hardness, the ultimate tensile strength f_{tu} was calculated from the formula

$$f_{tu} = \frac{1{,}000 \times 3{,}750}{130 - R_b}, \text{ psi}$$
$$= 45{,}500 \text{ psi}$$

The results of four runs are given in Table I. Speed of unloaded flywheel was 134 rpm. For lubrication, Sovarex L-1 was used for the flywheel bearing, and Vactra B-B for the ball screw, main and throw bearings. Before the test, the press was run about 15 min. at 200 rpm to wear in the lubricant.

Run No. XV was selected for detailed analysis. A photostatic enlargement of the trace for this run was made to assist in reading data accurately. Tonnage at different crank angles for this run are shown in Fig. 3. Fig. 4 is a plot of crank angle versus time and rpm versus time for the run.

To determine the friction loss for Run No. XV, it was necessary to find the total energy given up by the flywheel, the power supplied by the motor during the working portion of the stroke and finally the energy delivered by the punch to blank out the sheet. The energy given up by the flywheel was determined by multiplying the moment of inertia of the flywheel by one half the difference in the squares of the free velocity and the lowest velocity in radians of the flywheel during the stroke.

The engine energy was found by assuming the engine horsepower to be constant throughout the speed range and multiplying that by the time of the working stroke.

The punch energy was determined by multiplying the punch load by the

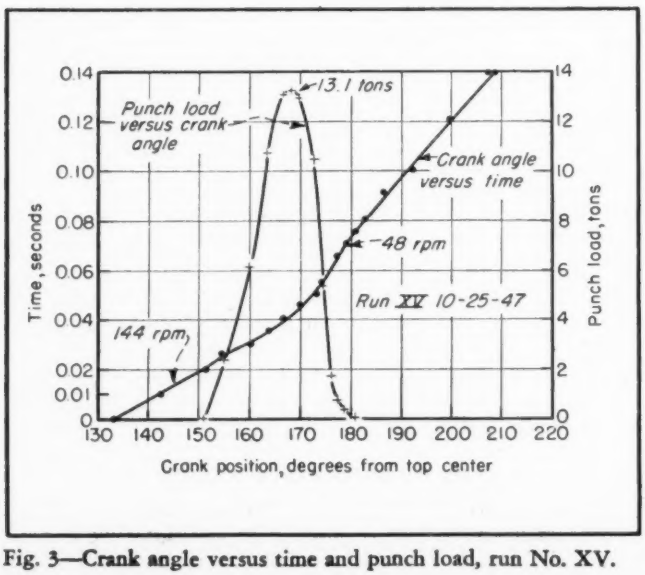


Fig. 3—Crank angle versus time and punch load, run No. XV.

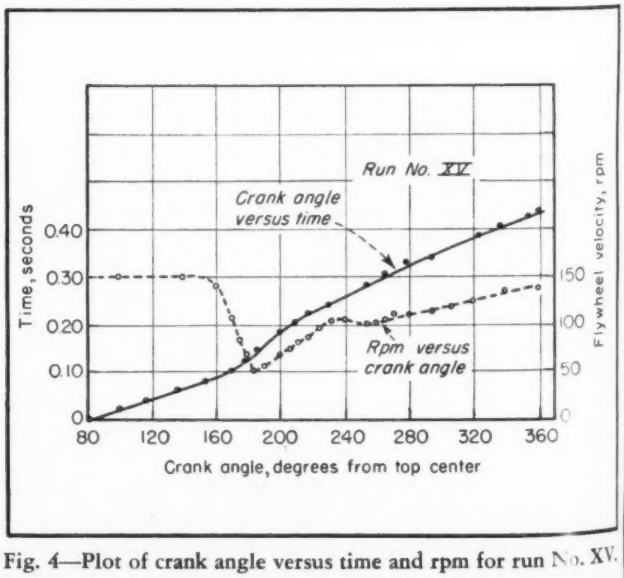


Fig. 4—Plot of crank angle versus time and rpm for run No. XV.

punch travel. Punch movement as a function of crank movement was found by measurement and also by calculation assuming the press frame to be infinitely rigid, Fig. 5. The plot of punch load versus punch travel, Fig. 6, was made from Figs. 3 and 5, for blanking out a piece of 0.108 in. hot rolled sheet. The data for Fig. 6 is presented in Table II.

The energy lost in friction in Run XV was calculated as follows:

Flywheel inertia = 2.5 slug-ft²
Initial flywheel velocity = 144 rpm
= 15.1 rad per sec
Lowest flywheel velocity = 48 rpm
= 5.03 rad per sec
Engine hp = $1\frac{1}{2}$
Time of work stroke = 0.046 sec
Energy given up by flywheel

$$= \frac{2.5(228 - 25.3)}{2} = 253.5 \text{ ft lb}$$

Energy supplied by engine
$= 1.5 \times 0.046 \times 550 = 38$ ft lb
Energy consumed by punch, from Fig. 6
= 135.1 ft lb
Therefore the percentage of available energy lost in friction was

$$\frac{253.5 + 38 - 135.1}{253.5 + 38} = 53.7 \text{ percent}$$

The apparent coefficient of friction in the frame and throw bearings, for the condition that all of the friction loss occurred in these bearings, was calculated from the following formula where

μ = average coefficient of friction in bearing
θ = crank angle, deg
P = average punch load over the range $\Delta\theta$, lb
r = radius of crank shaft, in.
A_f = projected bearing area of one frame bearing, sq in.
A_b = total area of both frame bearings plus throw bearing, sq in.
I_p = polar moment of inertia of flywheel, slug ft²
w_1 = flywheel velocity at θ_1, rad per sec
w_2 = flywheel velocity at θ_2, rad per sec
H = average brake hp of driving engine available between w_1 and w_2, hp
S = flywheel slowdown
$= (w_1 - w_2)/w_1$
W_p = work performed by punch, ft lb
t = time, sec
$= t_2 - t_1 = \Delta t$

$$\mu = \frac{875\, A_f}{P A_b r\, \Delta\theta} \times \left[\frac{I_p w_1^2 S(2 - S)}{2} + 550\, H\, \Delta t - W_p\right] \quad (1)$$

In applying this formula to run No. XV, it was assumed that: (1) Punch load versus crank angle movement was parabolic, therefore the average punch load equaled 0.666 times the maximum punch load, (2) Engine horsepower of $1\frac{1}{2}$ was constant, (3) The friction loss per bearing was proportional directly to the bearing area. Evaluation of the formula gave μ equal to 0.083 for run No. XV average.

For the die under test, it can be seen from Fig. 6 that the configuration of the plot of punch load versus punch travel can be approximated by a triangle. Assuming that the plot of punch load versus travel will be approximately the same for all thickness of metal blanked on this die, it is possible to rewrite Eq (1) so that an expression relating maximum punch load P_{max} and flywheel slowdown S is obtained, from which a theoretical plot of P_{max} versus S can be drawn.

Where

P_{max} = max punch load, lb
D = shear of die, in. For this case D = 0.09 in.
p = penetration of blank, in. For this case p = 0.30 in.
h = thickness of blank, in.

Assume all runs made at 144 rpm and that

$$W_p = \frac{P_{max}(D + ph)}{2 \times 12} \text{ ft lb}$$

$D + ph$ = constant = 0.123 in. for this case
μ = 0.083 average
t = constant for all values of P; = 0.046 sec

Then for a die of the type used on run XV

$$P = \frac{875 A_f [I_p w_1^2 S(2-S)/2 + 550 H \Delta t]}{\mu A_b r \Delta\theta + 875 A_f [1.5(D+ph)/24]} \quad (2)$$

For the 14 ton punch press and the die used on run XV at 144 rpm

$$P = 59.7\,[285S\,(2 - S) + 38] \quad (3)$$

or

$$P_{max} = 89.7\,[285S\,(2 - S) + 38] \quad (4)$$

From Eqs (3) and (4) it is possible to predict the relationship between maximum punch tonnage and flywheel

Table I—Summary of Blanking Test Data

Run	Dimensions of Hot Rolled Sheet, in.	Ball Screw Strain, micro in./in.		Punch Load Max, tons	Lowest Speed of Flywheel, rpm	Flywheel Slowdown, percent
		Back	Front			
XII	0.068 × full die	−367	−206	5.8	112	16
XIII	0.108 × 5	−831	−426	12.6	55	59
XIV	0.108 × 5.25	−938	−483	14.3	10	92
XV	0.108 × 5.125	−855	−440	13.0	48	64

Table II—Load and Punch Travel, Run No. XV

Crank Angle, deg.	θ, deg.	Punch Travel, in.	Punch Load, tons
151.0	29.0	0	0
155.0	25.0	0.0332	2.4
160.0	20.0	0.0681	6.1
163.8	16.2	0.0896	10.7
167.0	13.0	0.1043	13.1
170.0	10.0	0.1151	12.9
173.0	7.0	0.1232	10.4
174.6	5.4	0.1265	5.4
176.2	3.8	0.1290	1.7
177.6	2.4	0.1303	0.73
179.0	1.0	0.1311	0.34
181.0	−1.0	0.1311	0

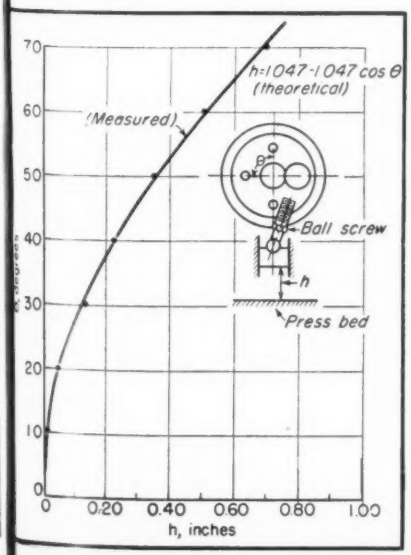


Fig. 5—Gap versus crank movement.

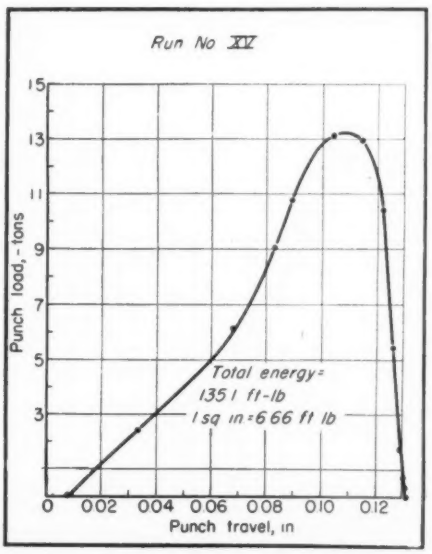


Fig. 6—Punch load versus punch travel.

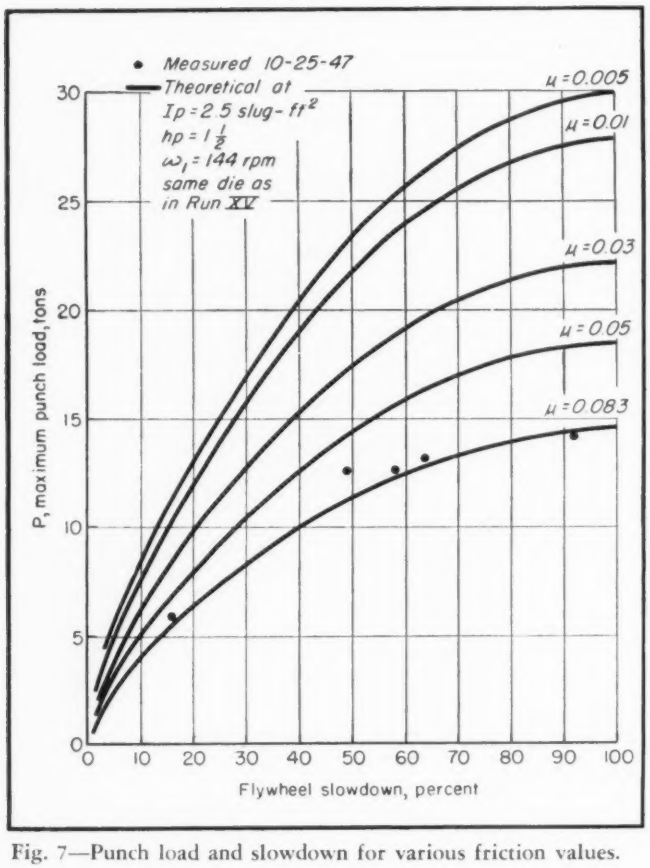


Fig. 7—Punch load and slowdown for various friction values.

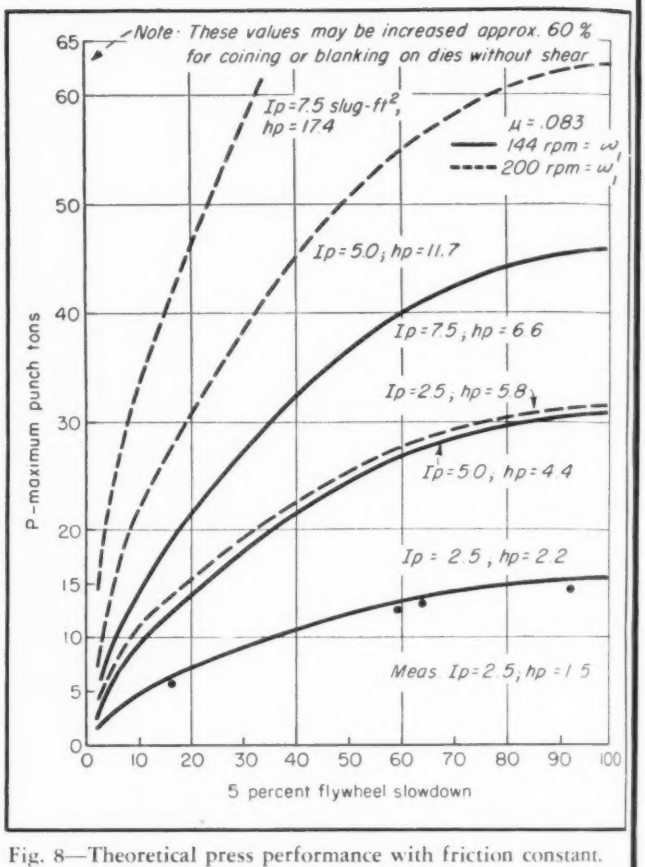

Note: These values may be increased approx. 60 % for coining or blanking on dies without shear
Ip = 7.5 slug-ft², hp = 17.4
μ = .083
144 rpm = ω₁
200 rpm = ω
Ip = 5.0; hp = 11.7
Ip = 7.5, hp = 6.6
Ip = 2.5, hp = 5.8
Ip = 5.0, hp = 4.4
Ip = 2.5, hp = 2.2
Meas. Ip = 2.5; hp = 1.5
P - maximum punch tons
5 percent flywheel slowdown

Fig. 8—Theoretical press performance with friction constant.

slowdown for this press and blanking die. Also, from Eqs (2), (3) and (4) it is possible to predict the performance of the 14 ton press using improved frame bearings.

For the 14 ton press running at 144 rpm, Eq (2) reduces to

$$P = \frac{5{,}910\,[285S\,(2 - S) + 38]}{430\mu + 30.3} \qquad (5)$$

The curves in Fig. 7 were plotted from Eq (5) using various values of bearing friction coefficient and slowdown. Since Eqs (2) to (5) do not correct for drop in engine horsepower as synchronous speed is approached, values of P are obtained that are non-existant at zero percent flywheel slowdown. In Fig. 7 these curves are drawn into 0 from about S equal 10 percent. The value of μ equal to 0.083 closely follows observed performance of the 14 ton press. The values of μ equal to 0.01 and 0.005 are for high quality automotive insert bearings and roller bearings respectively. (See "Internal Combustion Engines," Streeter and Lichty, 4th Edition.)

From an analysis of run No. XV it may be assumed, with reasonable accuracy, that the torque developed by the 1½ hp, 3 phase squirrel cage motor, used on the press, is essentially constant from about 40 to 100 percent full speed.

For simplification it is further assumed that the engine torque is constant over its entire speed range. On this basis equations can be derived relating engine horsepower, flywheel inertia and free rpm with slowdown and punch load, from which the effects of varying H, I_p on S and P, for a given value of μ can be studied. Where

a = angular acceleration, radians per sec per sec
T = engine torque, ft lb
t = time, sec
w_1 = free speed of flywheel, radians per sec
w_2 = lowest speed of flywheel, radians per sec
θ = angular movement of flywheel, radians
S = percent flywheel slowdown divided by 100
H = maximum brake hp of engine
I_p = flywheel polar moment of inertia slug ft²

$a = T/I_p$

$w = at + w_2 = Tt/I_p + w$

$$\theta = \int_0^t (Tt/I_p + w_2)\,dt = Tt^2/2I_p + w_2 t$$

and since

$$t = \frac{w_1\,(w_1 - w_2)\,I_p}{w_1 T} = w_1 SI_p/T$$

Table III—Horsepower Required to Restore Flywheel Speed

I_p	w_1 = 144 rpm		w_1 = 200 rpm	
	S = 1.0	S = 0.64	S = 1.0	S = 0.64
2.5	2.5	2.2	6.7	5.8
5.0	5.0	4.4	13.4	11.7
7.5	7.5	6.6	20.0	17.4

Table IV—Equations For Theoretical Press Performance

I_p	At 144 rpm	At 200 rpm
2.5	$P = 89.7\,[285S\,(2 - S) + 55.7]$	$P = 89.7\,[\ 550S\,(2 - S) + 147]$
5.0	$P = 89.7\,[570S\,(2 - S) + 111]$	$P = 89.7\,[1{,}100S\,(2 - S) + 296]$
7.0	$P = 89.7\,[855S\,(2 - S) + 167]$	$P = 89.7\,[1{,}650S\,(2 - S) + 440]$

and T equals $550\,H/w_1$, which is assumed to be constant
therefore

$$\theta = \frac{w_1^3\, I_p\, S\,(2 - S)}{1{,}100\, H}$$

It may be assumed that the slowest speed of the flywheel is reached at a crank angle of 180 deg or bottom center. The flywheel, therefore, must be accelerated back to full speed in the remaining 180 deg before top center. The horsepower required to bring the flywheel back to speed from any value of S at bottom center is

$$H = \frac{I_p\, w_1^3\, S\,(2 - S)}{1{,}100\, \pi}$$

In Table III values, calculated from Eq (6), are given for the horsepower required to restore flywheel speed from various values of S at bottom center as a function of I_p and free rpm w_1. Assuming that it is desirable to have sufficient engine horsepower to restore free rpm to the flywheel in 180 deg from a 64 percent slowdown, Eqs (2) and (6) can be combined to predict the performance of the 14 ton press and the same small blanking die with variations in engine horsepower, rpm and flywheel inertia, at a constant value of μ equal to 0.083. In Table IV equations are given for these variations. The curves in Fig. 8 were drawn from these equations.

By plotting the maximum punch load versus flywheel slowdown, Table I, on Figs. 7 and 8, it can be seen that the equations here presented for predicting press performance are in reasonable agreement with observed data.

It is of interest to predict the performance of the 14 ton press if its frame and throw bearings could be improved. Referring to Fig. 7, it can be seen that if the coefficient of friction for the bearings could be reduced from 0.083 to 0.005, the press would develop twice the punch tonnage for the same value of slowdown, and yet the engine, flywheel and initial rpm could be the same. According to Streeter and Lichty, high quality insert type bearings can have values of μ as low as 0.002, comparable to good roller bearings.

A study of Figs. 7 and 8 shows that the performance of the 14 ton press can be doubled by three different methods:

1. Reducing the friction coefficient to 0.005, without changing the rpm, flywheel inertia or engine horsepower.
2. Doubling the flywheel inertia and increasing the engine horsepower from $1\frac{1}{2}$ to $4\frac{1}{2}$, without changing the initial rpm or the friction coefficient.
3. Increasing the horsepower of the motor to 5.8 and the initial press speed to 200 rpm, without changing the friction coefficient or flywheel inertia.

These three methods supply sufficient horsepower to restore the flywheel to initial rpm from 64 percent slowdown at bottom center.

The most efficient method of doubling press performance, of course, is to reduce the overall coefficient of friction in the bearings.

It must be emphasized that all of the charts here presented are based on the use of the small blanking die, Fig. 2, but that the basic Eq (1) can be modified for any type of die by calculating the energy requirements of the work. Energy required for blanking on dies with or without shear can be estimated by the method used in deriving Eq (2). Energy required for drawing operations is discussed by G. Brewer and M. Rockwell in "Stress Strain Relationships in the Drawing of Metals," *Metal Progress*, June 1942, and by E. V. Crane in "Plastic Working in Presses," 3rd edition.

According to E. V. Crane, past practice was to add a "liberal" percentage to flywheel and engine energy calculations to cover machine friction.

During the runs reported here, when checked with a counter and stop watch, the free rpm of the flywheel was found to 134. Run No. VII, made with the oscillograph, indicates an average velocity of about 134 rpm, but apparently shows a speed of 145 rpm at 90 deg and 125 rpm at 340 deg. Whether the press, with the ram operating, actually does fluctuate in velocity over the stroke, or whether this velocity change is not real but results from some peculiarity of the recording system is not known. For this reason, all calculations based on the oscillograph recordings are for an initial 144 rpm entering the working stroke.

High Pressure Vessels Designed with Wound Construction

WAR PRODUCTION of synthetic oil was speeded in Germany by designing high pressure vessels to be made by winding metalic tape on thin-walled tubes. This was one of three outstanding features in the engineering of German hydrogenation plants according to a recent Office of Technical Services report prepared by J. F. Ellis of the British Intelligence Objectives Subcommittee. Automatic control of hydrogenation units and development of new steel were the other two features.

Metallic tape is wound on thin-walled tubes until there is sufficient depth to provide adequate strength up to 4,000 atmospheres. Tape is either 5 by 50 mm or 8 by 80 mm, according to size and working pressure of the vessel. Compositions and details of manufacture of tapes are included.

Vessels operating at temperatures up to 200 C are wound with a normal carbon steel having a carbon content of 0.20 percent and a yield point of 23 tons per sq in. For vessels up to 350 C, a steel tape, with 0.15 carbon, 1.0 chromium, 0.5 percent molybdenum, and a yield strength of 22 tons per sq in. at 350 C, is used.

The lathe on which the vessel is wound is of normal design used for machining solid wall vessels of the same size. The tool carriage is altered to tow a large frame. The frame runs on a special track and carries a large spool of tape, as well as some of the tape heating equipment.

In developing wound vessels, much additional information was discovered concerning design of ordinary high pressure vessels. The main changes in the design of forged vessels are in the reduction in weight of end covers and end joints. A table and several drawings illustrate progress made. General design of gas-fired preheaters and types of steel used in their construction are discussed at length. German experience with a 700 atmosphere (10,500 psi) hydrogenation plant suggest that they have not found a steel entirely suitable for use at such a pressure and at a metal temperature of 550 C. Preheaters using the developed steel at temperatures of 500 C had creep strength of 15 tons per square inch.

Two addenda reports contain information on results to date of tests on hydrogen attack on unstressed test rods as a function of hydrogen pressure, time of test and state of heat-treatment of material, and considerations of the design of hollow cylinders for high internal pressure at high temperature and recommendations for a more realistic basis of calculation.

Mimeographed copies of the 152-page report (PB-80349: "Engineering in Hydrogenation Plants in Germany") cost $4.25. Orders should be addressed to the Office of Technical Services, Department of Commerce, Washington 25, D.C., and should be accompanied by check or money order made payable to the Treasurer of the United States.

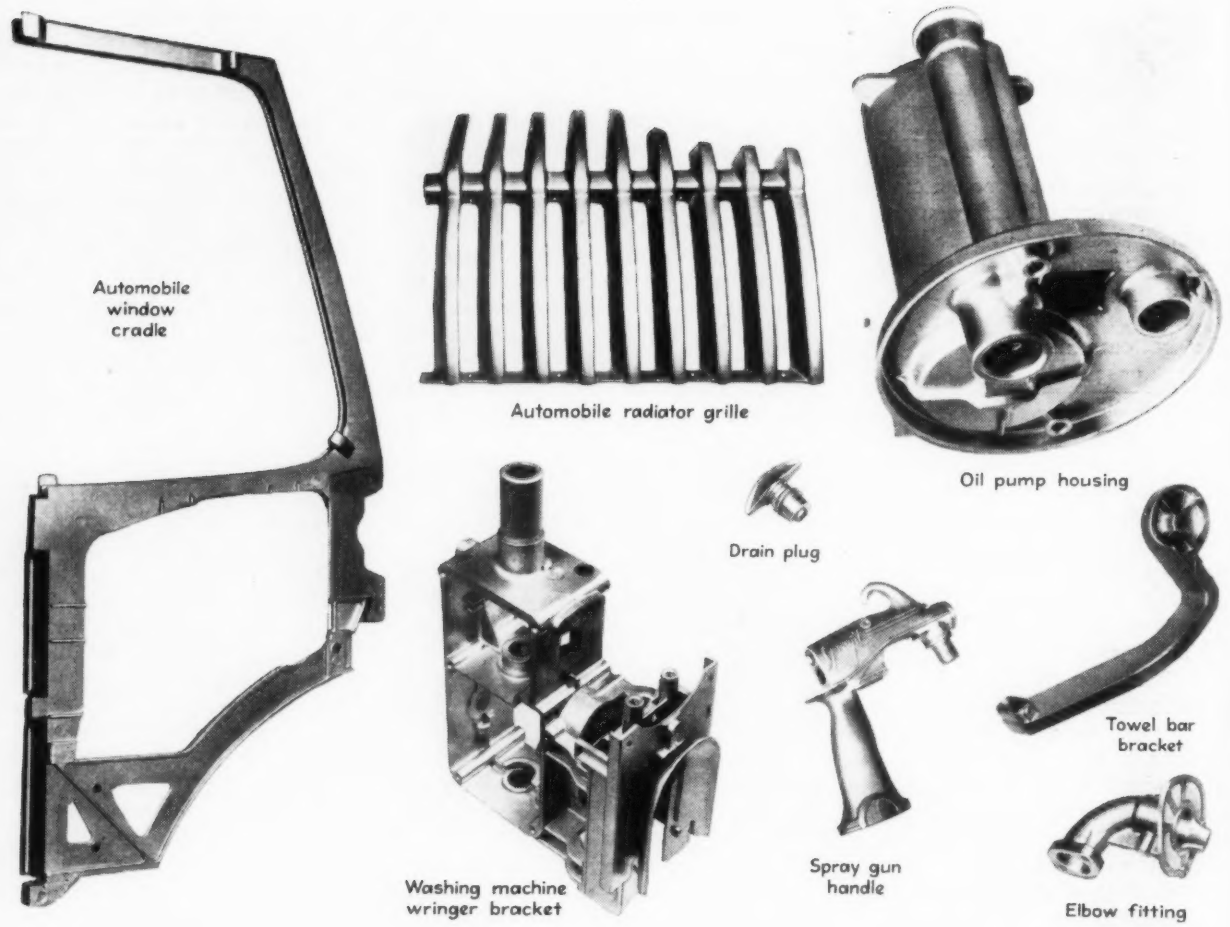


Fig. 1—Wide range of shapes and sizes possible allows the die casting process to be applied to many different products

Designing for Die Casting

Important factors in designing for die casting include choice of material, size, shape, dimensional tolerances and pressure requirements. An understanding of die design is necessary, since cost and ease of production depend on the shape of the part.

H. C. SNYDER, B. E. SANDELL, L. E. CAPEK, G. NIELSEN

Metallurgical, Engineering and Operating Departments
Stewart Die Casting Div., Stewart-Warner Corporation

DESIGN ADVANTAGES of die cast parts stem from the inherent characteristics of the process. Basically, die casting differs from other casting methods in three respects:

1. A permanent die is used—one that is not destroyed in each casting cycle. Even the cores are permanent, unlike "semi-permanent" molds.

2. The metal is molten—not merely "plastic" as in extrusion.

3. The pressure is comparatively high—unlike other "permanent mold" processes.

Because of these factors, there are numerous advantages in the finished product. First, little or no machining is required on the die cast part. There are several reasons for this:

1. Castings can be made to close dimensional tolerances.

2. Finished surfaces are smooth—this also improves physical properties.

3. Surface details, such as external threads and decorative designs, can be included in the die.

4. Internal details, such as threads or bushings, are possible.

Final assembly operations are simplified or made unnecessary, because the castings are produced uniformly and accurately. Special-purpose requirements are met by the use of inserts—bosses, panels, plugs or other reinforcements can be included in the casting.

Pressures used in die casting make for high mechanical strength in the finished part. Thinner wall sections are possible, resulting in reduction of overall part weight.

General Design Requirements

Die casting permits a flexibility of design that is lacking in many other methods. Before full advantage can be taken of this process major design factors should be understood.

MATERIALS. Only alloys with comparatively low melting points can be used. This restricts the choice of metals to: Tin; lead and tin-lead alloys; zinc alloys; aluminum and magnesium alloys; and low melting point brass alloys.

It should be determined whether the particular application can be made from one of the available die casting alloys; physical and chemical properties of the material chosen should be carefully studied. Tin-base alloys are frequently used where absence of contamination in food products is important—such as in soda fountain parts. Lead-base alloys are employed in printing industry applications, where they resist corosive action of ink, and for many bearing uses. Zinc-base materials are the most widely used in die casting—representative applications include automotive parts, home appliances and business machines.

Aluminum alloys are selected for die casting when weight economy is important, and where certain types of corrosion resistance is important, as in cooking utensils. Magnesium-base alloys are considered mainly for weight savings. Magnesium alloys, however, have certain physical properties, such as high yield strength, which also govern their use. A typical application where this property is important is an aircraft landing gear. Copper-base alloys are least used in die casting because of short die life. However, they resist certain types of corrosion, and exhibit better physical properties than any of the other die casting materials.

SIZE. Die casting at this time is generally limited to a 25 lb casting in zinc, or its volume equivalent in other materials. As shown in Fig. —, size possibilities are unlimited within this weight range.

SHAPE. Form of the part should be as simple as its functional and structural requirements will permit. This will allow use of the most easily constructed die, speed production, and facilitate assembly and finishing operations. As for the casting operation itself, there are two main considerations: Undercuts and draft.

Most die castings are made in separating dies, consisting of two halves, with a common parting line that is determined by the design of the part. Surfaces containing undercuts prevent free withdrawal of the part from the die cavity, increase die cost, slow casting, and result in flash which necessitates a subsequent finishing operation. They should be avoided, if possible, by redesign. A water faucet, for example, could not be cast in a separating die. Such parts are usually sand cast; the sand cores are easily removed.

Certain parts that contain undercuts can be cast by using loose pieces, called "slides" that eject freely with the casting (Fig. 2AB). With a special fixture, these are subsequently removed from the part—but not in the plane of withdrawal of the part from the die. If a hole is required in the finished part, the loose piece is called a "core;" core holes can be casd in any plane or side of the piece. Both slides and cores are made of the same material as the die, and are considered as permanent parts of the die.

Free ejection of the part from the die requires that sufficient draft be allowed. As shown in Table I, this makes impossible casting of parts having extremely narrow draft limits. For example, a cylindrical part having a draft of only 0.001 in. per in. cannot be cast. If draft interferes with the final design of a part, it can be subsequently machined to the proper shape.

Reverse draft in holes or undercuts should be avoided. This prevents withdrawal of cores and slides in a straight line. Holes with reverse draft can be made by using collapsible cores, but these are usually expensive.

DIMENSIONS AND TOLERANCES. Thickness of wall sections should be as uniform as the design of the part will permit. When change in thickness of section is necessary, transition from thin to heavy sections should be as gradual as possible. This minimizes stresses set up because of unequal cooling rates, and avoids possible warping, cracking and distortion. Abrupt changes in section thickness are obtained only at the expense of solidity of structure, and should be avoided, or the part redesigned if possible.

It is usually desirable to keep wall sections thin, because addition of weight does not result in a proportionate increase in strength. Also wall sections that are too heavy may cause porosity, and prevent a solid structure from forming.

However, wall sections that are too thin restrict flow of metal into the die. Minimum wall thicknesses recommended for die cast parts are shown in Table I. Applying the table, it can be seen that it is difficult to cast products which, for example, were previously fabricated from sheet materials less than 0.050 in. thick. Similarly, it is difficult to obtain a dense structure in a part, such as a large diameter wheel, about 1 in. thick.

Dimensional tolerances required in the finished part should be within the

Table I—Approximate Dimensional Limits for Die Castings

	Zinc Alloy[1]	Aluminum and Magnesium alloys	Copper alloy (brass)
Minimum wall thickness, large castings, in.	1/16	0.085	1/8
Minimum wall thickness, small castings, in.	0.030	0.050	0.050
Tolerances in solid die section[2]			
On any one dimension, in. per in.	0.001	0.0015	0.003
Minimum on any one dimension, in.	0.0025	0.003	0.005
Cast threads, external, max. number per in.	24	20[3]	10
Cast threads, internal, max. number per in.	24[4]	none[3]	none
Cored holes, min. dia., in.	0.030	3/32	3/16
Minimum allowable draft, in. per in. of core length	0.003	0.015	0.020
Minimum allowable draft, in. per in. of cavity depth	0.005	0.010	0.020

[1] Tin and lead alloys are die cast to approximately the same dimensional limits as zinc alloys.
[2] Applies only to dimensions across solid portions of die. When dimension is across parting line or between points controlled by parts of die having relative motion, wider limits must be allowed.
[3] Small aluminum die castings having walls only 0.040 in. thick and threads as fine as 24 per in. have been produced in quantities.
[4] If cheaper than tapping. Fine internal threads are nearly always tapped.

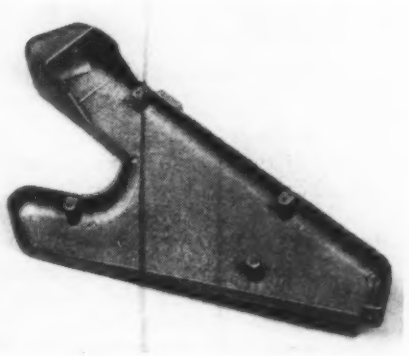


Fig. 2—Undercut in mimeograph case cover is cast with loose-piece core.

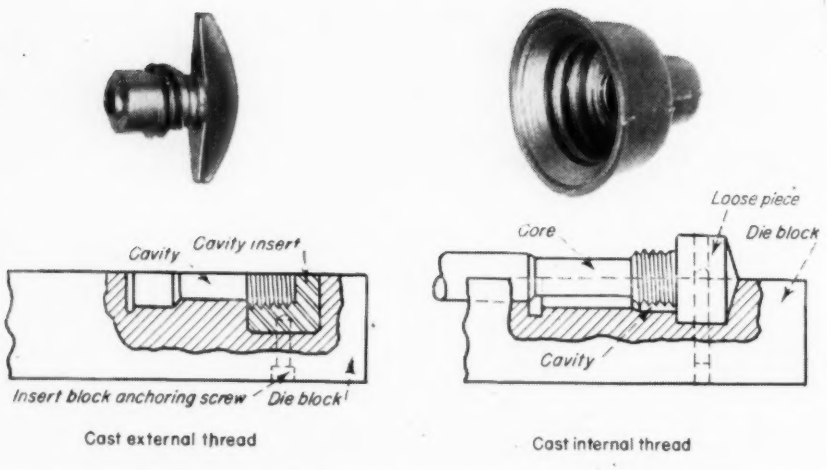


Fig. 3—External and internal threads are die cast in drain plug and soap dispenser.

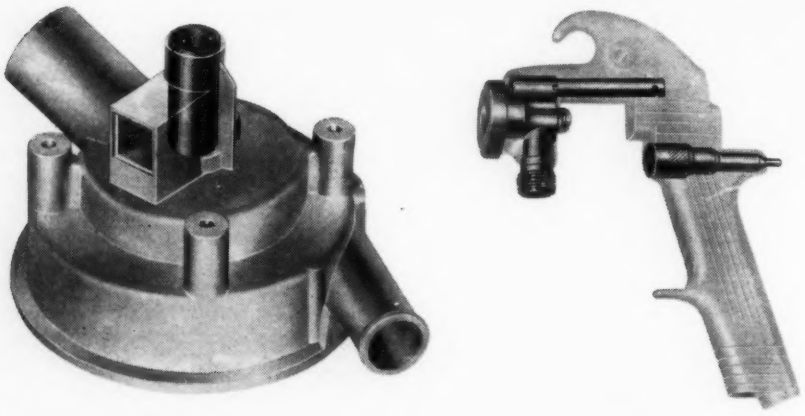


Fig. 4—Pump bracket and spray gun handle are die cast around metallic inserts.

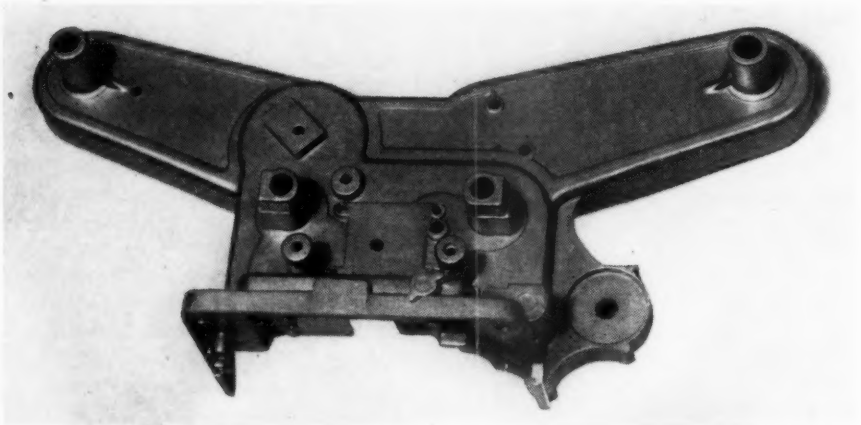


Fig. 5—Studs and bosses are cast in projector reel frame, for drilled and tapped holes.

practical limits of die casting, if this process is to be used. Tolerances indicated in Table I should be followed.

PRESSURE REQUIREMENTS. Designers should determine whether the finished part must withstand high liquid or gas pressures without leaking. Die castings are not recommended for this type of service because of the comparatively porous structure that is apt to appear in heavy sections. These relatively open grain structures result from internal shrinkage occuring during the die casting process. For example, die castings should not be used in a high pressure steam valve, or in hydraulic fittings where continuous high liquid pressures are developed.

QUANTITIES REQUIRED. For die casting, quantities required should be sufficiently large to warrant the construction of a die. Die casting dies are considerably more expensive than sand or "permenant" molds. If quantities are not sufficiently large to amortize the initial die cost, without placing a prohibitive price on the finished part, another method of production should be considered.

Special Design Details

THREADS. External and internal threads can be die cast. External threads should be fairly coarse, and should have pitch diameters not less than $\frac{3}{4}$ in. Fine threads of small pitch diameters are not easily cast. Their location and design should not interfere with normal parting of the die. When external threads are cast, flash will appear at the parting line, especially if the threads are cast on a core. This will necessitate a trimming operation when the casting is cleaned. Loose inserts that are subsequently screwed off an external thread to avoid flash are impractical; their use slows the casting process. Fig. 3 illustrates cast threads normally used in die casting.

Internal threads, Fig. 3, can be cast, but they require the use of extremely expensive dies, incorporating a threadout core, and the casting operation becomes slow. It is much more economical to machine internal threads into a part after casting, unless the shape of a part is so irregular that chucking or fixture mounting for a machining operation is unreasonably difficult.

INSERTS. Inserts are desirable for these requirements in the finished casting:

1. Added strength or stiffness.
2. Extra hardness at points of wear and specialized use.
3. Good bearing qualities.
4. Electrical or heat insulating properties.
5. Passages for liquids or gases.
6. Resiliency.
7. Special decoration.
8. Resistance to local corrosive agents.
9. Spring qualities.
10. Special cutting characteristics.
11. Soldered connections.
12. Simplification of intricate assemblies.

Inserts are usually made of materials dissimilar to the die casting alloy. Thus, physical characteristics not inherent in the die casting alloy can be designed into the finished casting by using suitable inserts. Metal inserts are most common, but some nonmetals are used. When inserts of a dissimilar metal are used, the possible

danger of electrolytic corrosion should be explored.

Because of the high shrinkage of die cast materials, the bond between insert and casting is usually strong, but it is good practice to strengthen the attachment by knurling, undercutting or grooving the anchor end of the insert. When designing an insert that must turn or have loose anchorage, without being removable, the insert is grooved and the anchor end is coated with a compound that prevents its bonding with the die cast metal.

When inserts are used the cost of the casting is usually increased, because of the cost of the insert, and placing the insert in the die slows the casting process. Also, allowances must be made for damaged inserts if they are not reclaimable. Fig. 4 illustrates typical insert applications.

BOSSES AND STUDS. Bosses usually furnish a sufficient mass of metal around a drilled or tapped hole to avoid local stresses. Bosses should not be too heavy, because of the added weight and because porosity tends to develop in heavy sections. This can prove objectionable when drilling, tapping or machine operations are required. If bosses are tapped, at least $\frac{1}{8}$ in. should be allowed at the bottom of the tapped hole for chip clearance.

Casting of studs which are to be threaded is not recommended, because threading produces a notch effect that weakens the structure. A boss that can be tapped is stronger and much more satisfactory. (See Fig. 5).

Bosses for drilled or tapped holes should be close to the desired finished dimension. Porosity in the casting is often found in massive sections. Accurate coring prevents the possibility of a tap, drill, or reamer striking a blowhole or void, which usually throws it out of alignment and results in tool breakage.

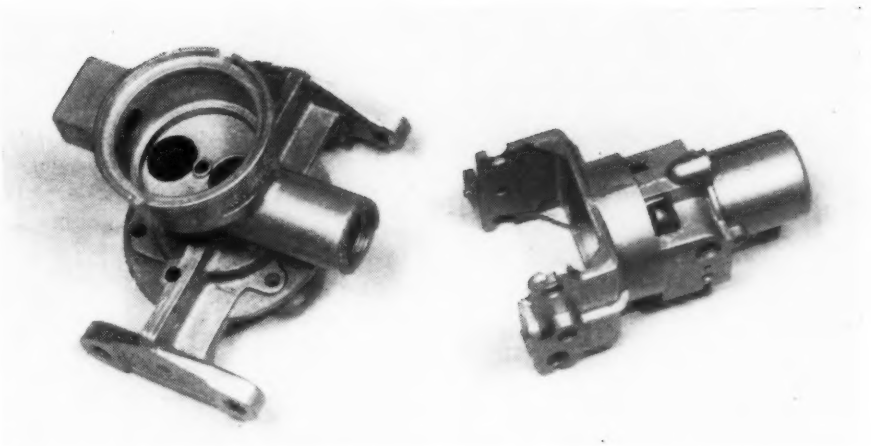


Fig. 7—Designs of fuel pump and speedometer frame require elaborate coring.

CORES. Cores should be of the simplest possible shape and should lie in planes parallel to that in which the die moves (Figs. 6 and 7). Cores lying in planes other than that of the die motion, require special equipment, which will be explained later. These additions to the die increase its cost and make its operation more intricate.

Regular cylindrical cores are the easiest to make and operate. If the hole must be other than cylindrical in shape, the end by which the core is fastened in the die should be cylindrical, so that the anchor in the die can be a simple drilled, or bored hole. This gives the best alignment, with a minimum of wear. Regardless of shape, cores should have sufficient taper to allow maximum ease of extraction. Cores that form blind rather than through holes are an advantage, because no flash is formed at the blind end and a trimming operation is avoided.

Extremely long holes, particularly if they are slender, can be the source of serious casting difficulties. It is recommended that cores under $\frac{1}{2}$ in. in diameter never be longer than three to four times their diameter. When the design permits, it is an advantage to extend the free end of a core into the opposite half of the die. This gives the core rigidity and better alignment, and prevents bending from the pressure of inrushing metal.

Intelligent use of cores has another advantage, in addition to producing functional holes. Sections of a die

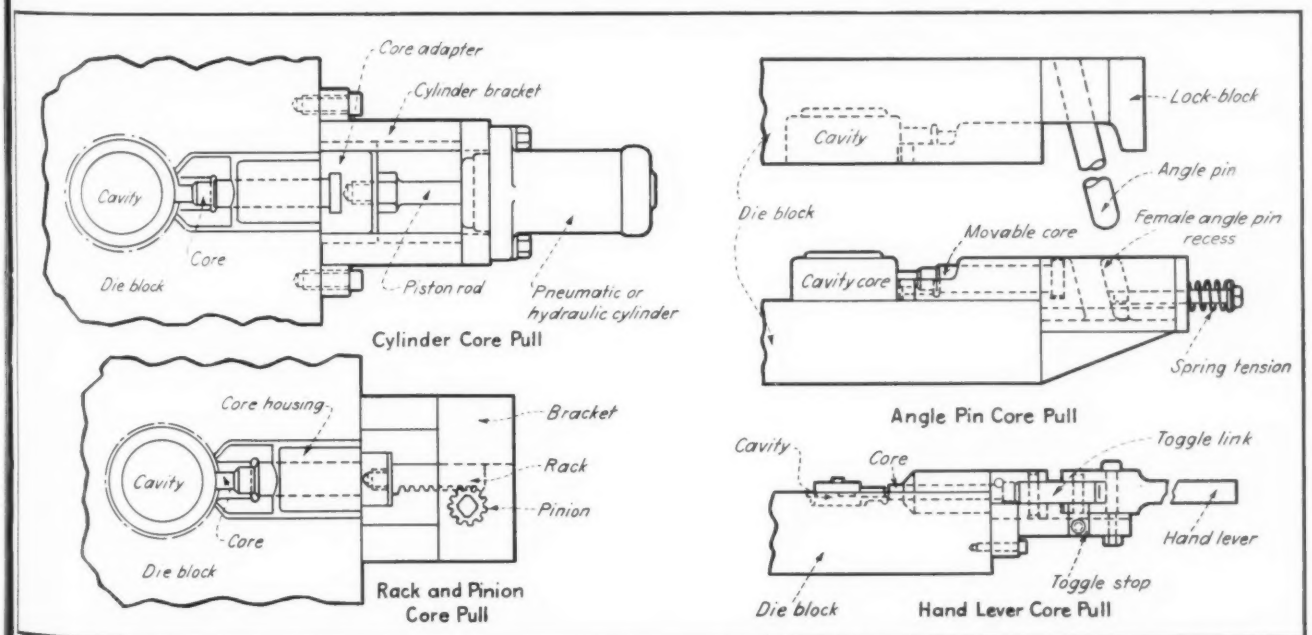


Fig. 6—Special core pulling mechanisms are necessary to cast undercuts and holes. When possible cores should lie in plane of die motion. Interlocking and thread-out cores should be avoided, since they slow casting and raise initial and operating costs.

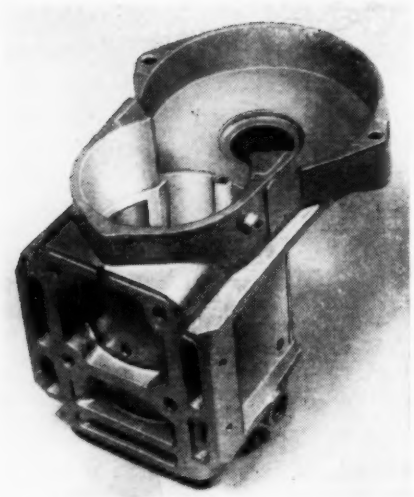


Fig. 8—Elongated slots in farm tractor magneto frame serve as metal savers.

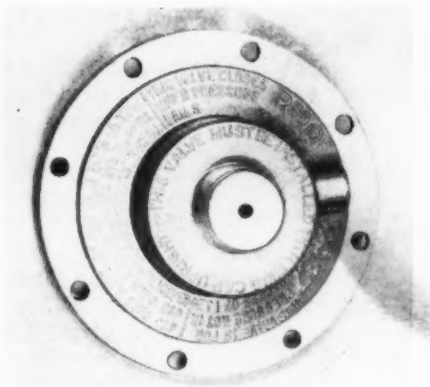


Fig. 9—Raised lettering is cast into valve bonnet for gas regulator assembly.

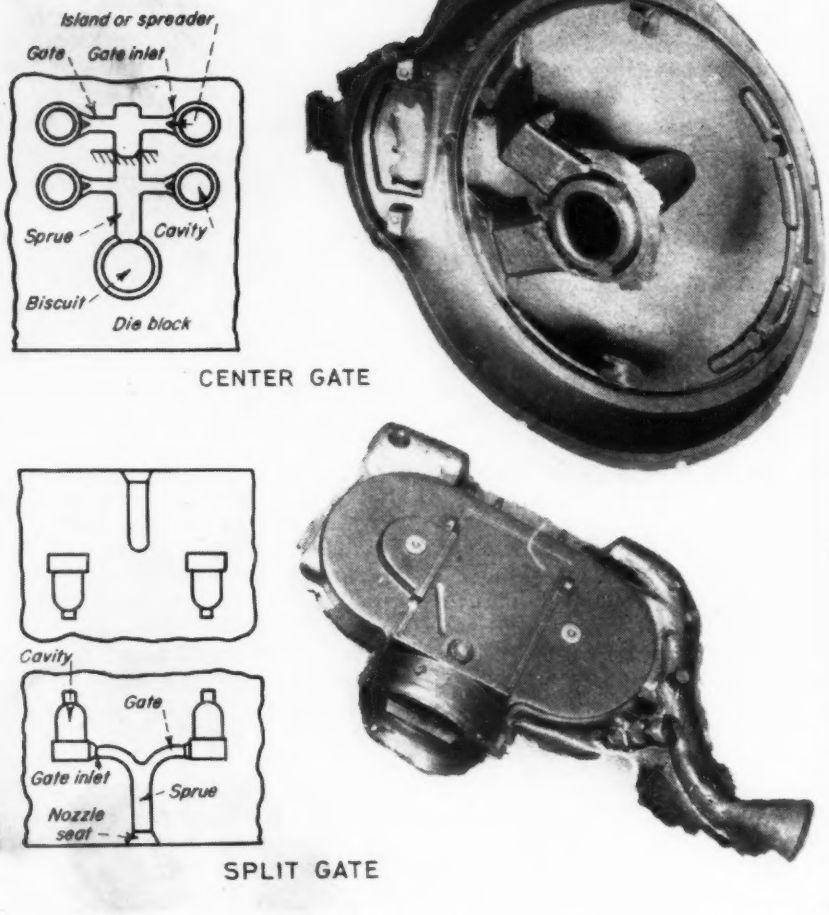


Fig. 10—Automobile head lamp rim (top) and 8 mm movie camera case (bottom) illustrate advantageous use of center and split gate die constructions.

cast part that might be unduly heavy, causing a tendency to porosity or representing an abrupt change from a thin section, can often be lightened through the use of cores or "metal-savers" (Fig. 8). Other advantages of such cores are overall weight economy and speedier production because of faster cooling.

SURFACE APPEARANCE. If appearance of the finished part is important, plain flat areas should be avoided whenever possible; these are difficult to cast without imperfections. Flat portions should be slightly curved or a simple decorative design should be incorporated. To avoid "shadow marks," which usually appear on a smooth surface when a rib, boss or other heavy section lies underneath, sections should be made 0.100 in. or more in thickness, and the underlying bosses or ribs should not exceed 80 percent of the thickness of the section. If this is impractical, the addition of a surface design will camouflage blemishes and other imperfections.

Identifications, instructions for use, or decorative designs, can all be included in the die so that they will appear on the surface of the finished casting (Fig. 9). When such elements are incorporated, however, they should be in raised characters. Depressed characters in the casting require raised characters in the die, making necessary a difficult and expensive engraving operation. If characters must be kept flush with the surrounding surface, it is usually possible to cast a depressed panel into the surface and to use raised lettering on the panel; the additional die expense is nominal.

CORNERS. Fillets should be used liberally. Sharp corners increase die cost, cause "drag" and impede metal flow, cause formation of "checking" which is harmful to the appearance of the casting, and retard ejection of the casting. A minimum radius of 0.015 in. is suggested where sharp corners are specified. Larger radii are desirable, if they are permissible.

Although it is not the purpose of this article to discuss details of the die casting process itself, it is felt that the engineer will be able to design a better, more economical product with some knowledge of the fundamentals of die construction.

Die Construction

There are essentially two types of die construction: The split-gate and the center-gate dies. In the split-gate die, the metal enters the cavity through a sprue where the two die halves part; in center-gate construction, the sprue or inlet passes through the stationary half of the die. Fig. 10 illustrates the two types of die construction.

Many die cast parts are so designed as to require cores or slides to erect their ultimate shape. Cored holes can be cast in any plane or side of the piece; cores can be fixed as a permanent part of the die, or they can be arranged as removable pieces, Fig. 11. However, cores lying in planes other than that of the die motion, require the use of cylinders, angle pins, cams, rack-and-pinion, or collets to effect extraction. Certain types of undercuts

could be cast by employing the use of loose pieces which are ejected with the die casting and are subsequently removed after the part is ejected from the die. Fig. 2 illustrates this design feature.

Cylinders used to activate cores or other moving parts in a die can be powered by pneumatic or hydraulic pressure. In the angle pin method, a male angle pin engages with a female counterpart. It is also possible to arrange a lever system for these movable parts, which can be manually or mechanically operated. Figs. 6 and 11 illustrate methods used for core and slide operation within a die.

When a minimum of flash and perfect matching of both halves of the die cast part are important in the design, collet dies are used. They also permit the casting of certain types of undercuts. The collet die, shown in Fig. 12, is actually "a die within a die," and is designed to prevent any misalignment at the parting line. This type of die is particularly useful for casting external threads, or similar protuberances. The angular ejection principle of the collet die permits both casting outer surface undercuts in a vertical plane, and ejection of the casting, as the parting faces of the die separate in opposite directions. Precision "tool-making" is necessary in building this type of die. There is practically no limit to the variety of design features that can be incorporated into a die cast part, when undercuts and reverse draft are eliminated.

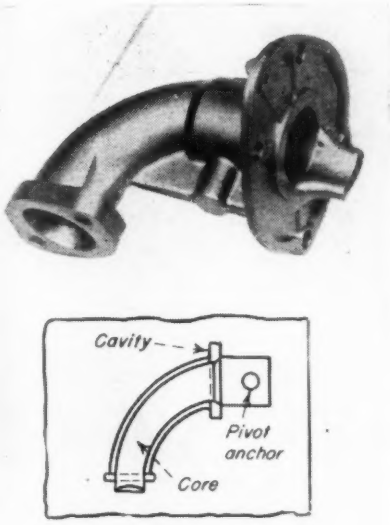


Fig. 11—Core for carburetor bowl elbow fitting is removed with casting.

EJECTORS. To facilitate removal of the casting, an ejection mechanism must be built into the die. This consists of a series of ejector pins mounted on plates, power-operated as the die opens, located in the back of the moveable half. Ejector pins should not be located so that they press against a section of the casting which might be too weak to withstand the ejection pressure, or where they might leave objectionable marks on the surface of the casting. Ejector pins are often placed where they press against ejector lugs which can be subsequently removed in the trimming operation. Shown in Fig. 13 is a typical part die cast with the aid of a cam-activated screw ejector—a lawn mower gear.

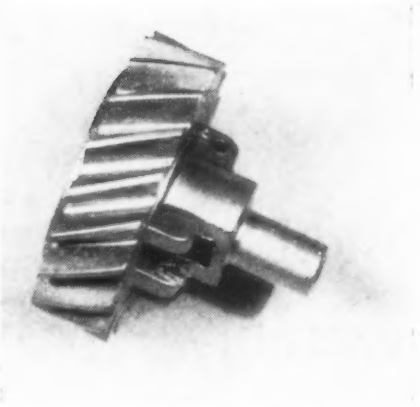
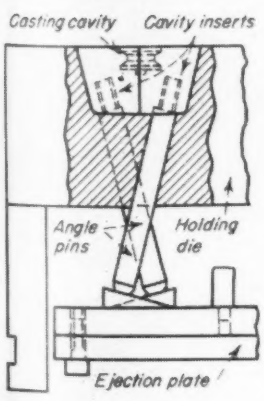


Fig. 13—Helical gear in power mower is cast with cam-activated screw ejector.

MULTIPLE-CAVITY DIES. When size and design permit, it is advantageous to construct the die so that two or more parts can be cast in each operation. Cavities required for each part are spaced around the sprue so that uniform metal injection is possible.

In designing a multiple cavity die care must be taken to avoid more cavity space than can be filled with a normal shot of metal. This makes it difficult to secure proper gating and may result in slowing of the casting operation. Identical or similar parts cast best in the same die; otherwise, it may be necessary to sacrifice structural perfection in one part to obtain a desirable feature in another.

In designing for die-casting, the product designer must work closely with the die designer and the die caster to obtain the best and lowest cost results from this valuable process.



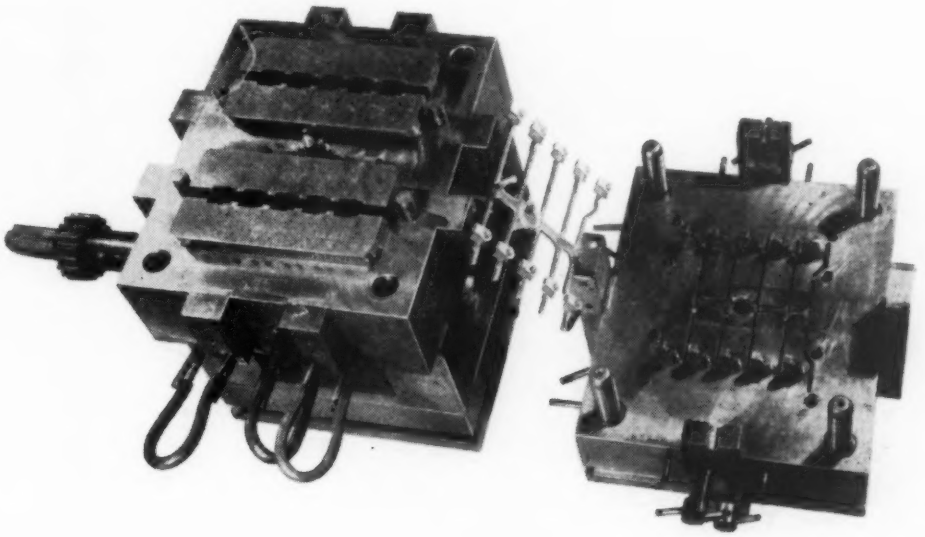
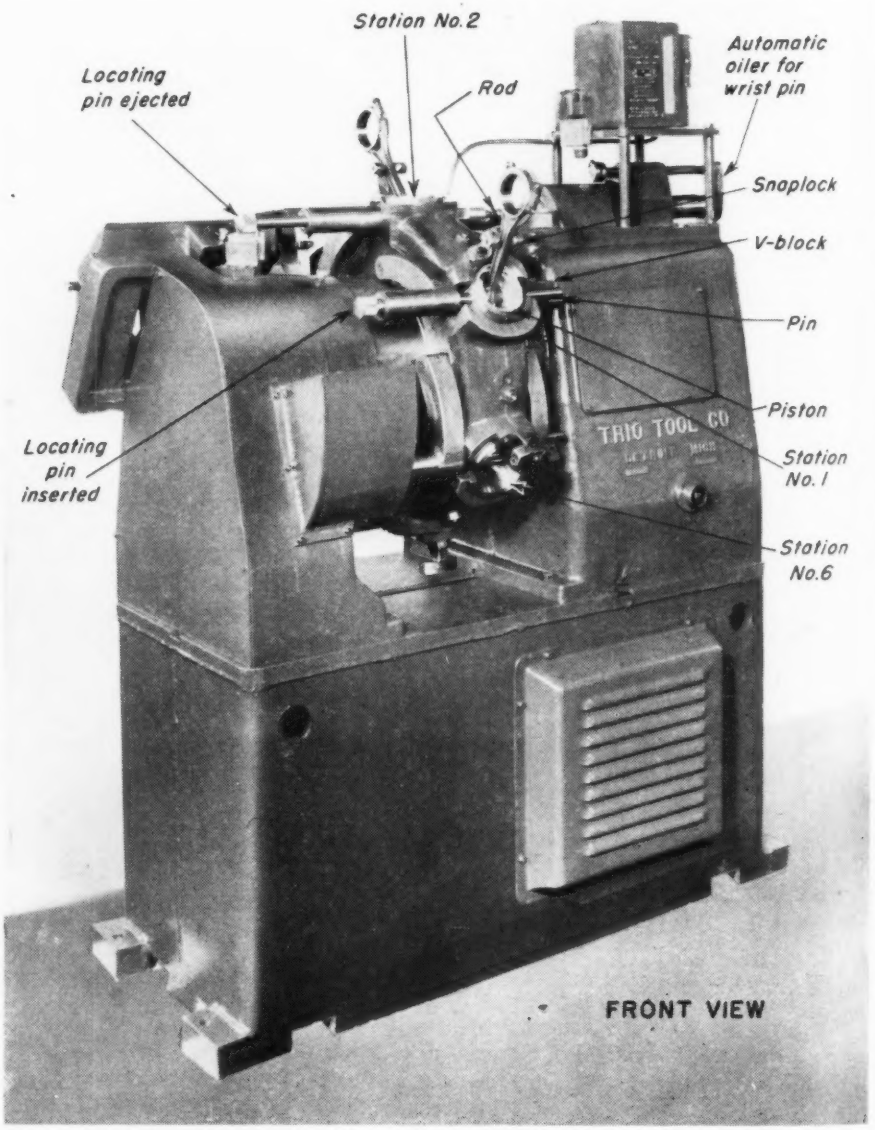
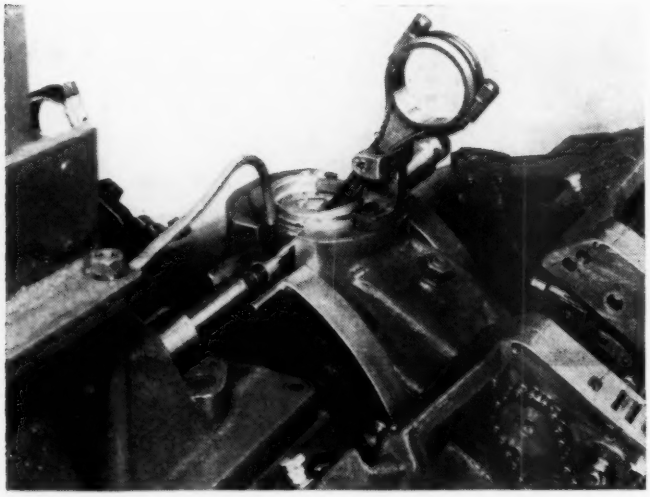


Fig. 12—Collet die, designed to prevent misalignment at the parting line, permits casting outer surface undercuts in a plane opposed to die motion. This type of die allows ejection of the casting as the cavity parting faces separate in angularly opposite directions.

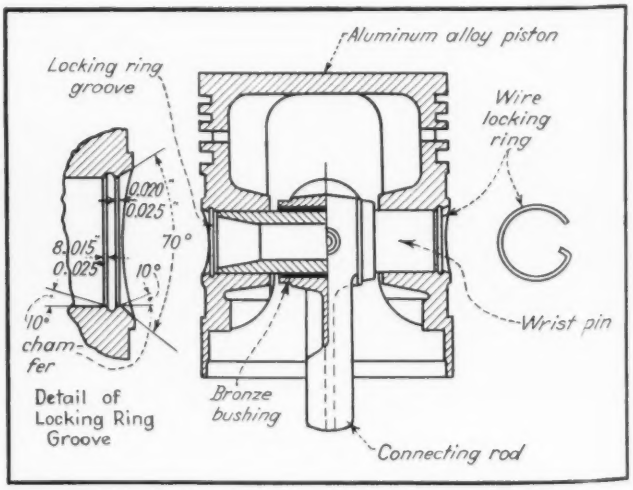
Piston Assembly Machine

Wristpins are automatically inserted in pistons and connecting rods by the assembly machine designed and built by Trio Tool Co. The machine also forms two snap rings and inserts them to retain the pin. Operator places the piston, rod and wristpin in one of six fixtures. Then he pushes locating pin through piston and rod. After indexing, the machine lubricates and inserts the wristpin to replace the locator, indexes again and inserts two snap rings. The completed assembly is ejected at the next station. Interlocks stop the machine if any step is not properly completed. A safety clutch protects the machine against overload. Machine may be operated continuously or in single cycles. Cycle time is 4.5 sec. First unit was installed at Ford Motor Co.

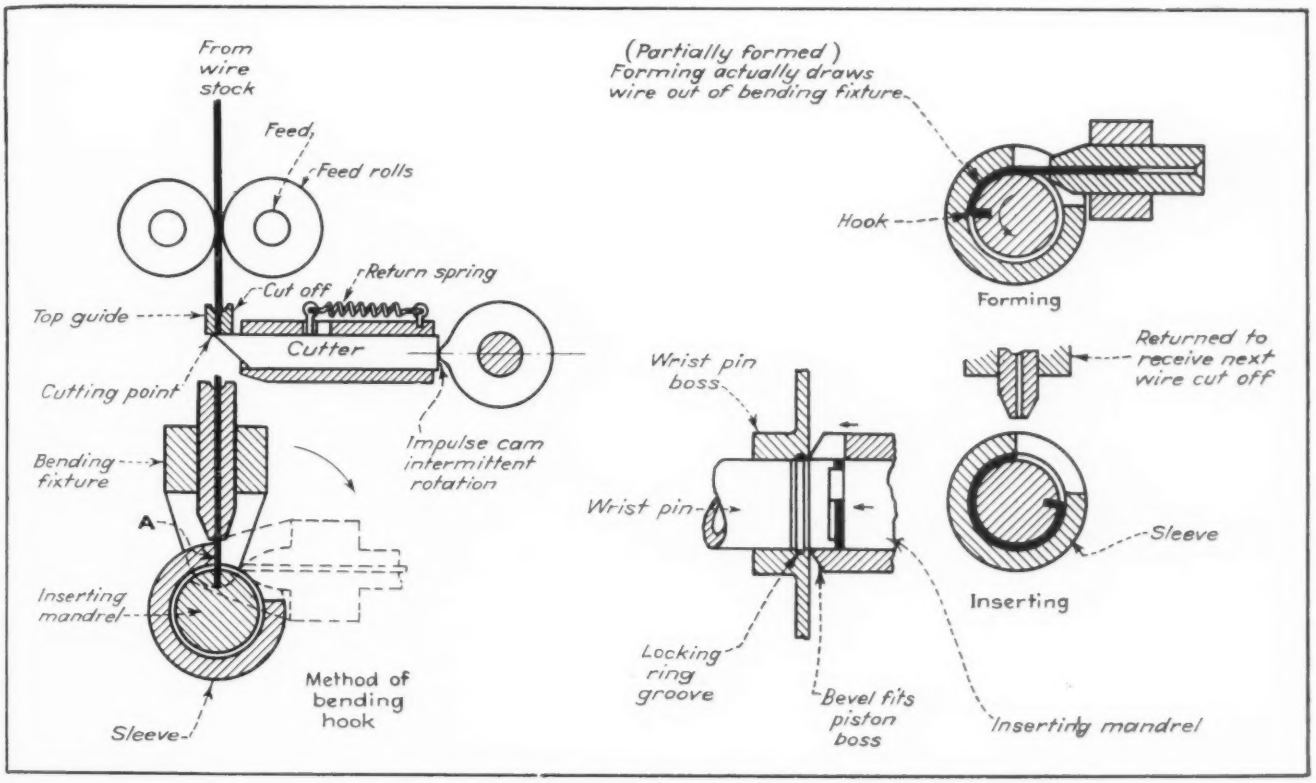


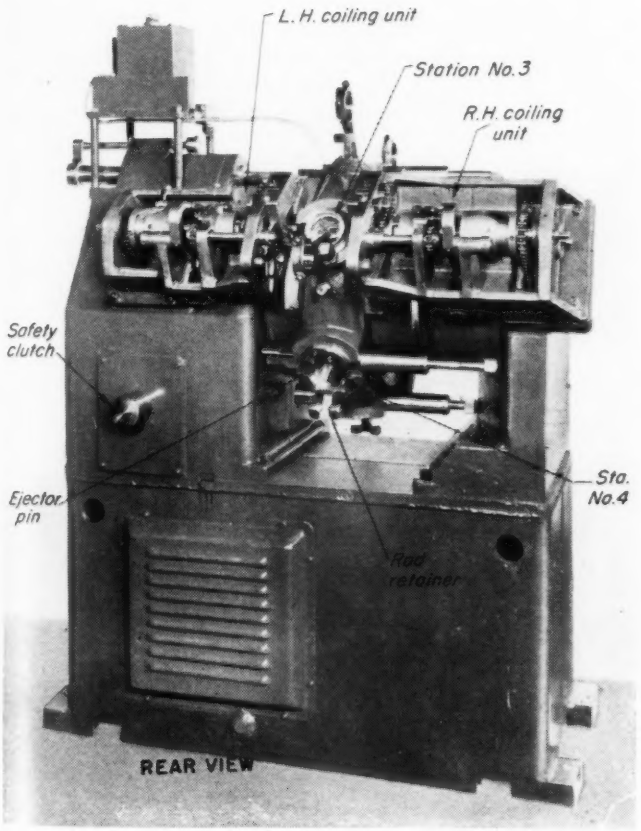


AT STATION 2, wrist pin is inserted. Each section of rotating wheel has vee block and locating pin for aligning wrist pin with piston and rod.



PISTONS are preheated to increase allowance for assembly. To prevent chilling, the chucks may be made with electric heaters.





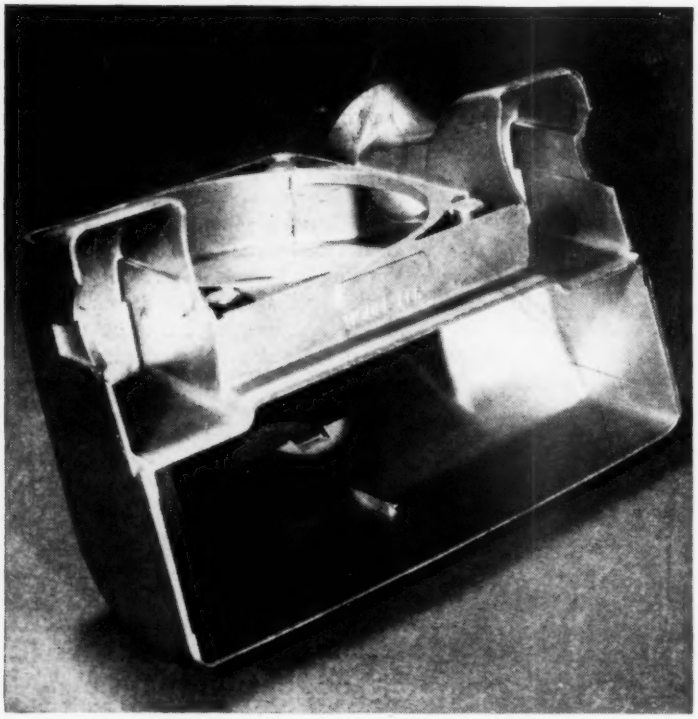
PISTON AND ROD ASSEMBLY is ejected at rear of machine by pin in bottom of chuck. Empty rod retainer at station 4 shows spring loaded holding balls.

RING FORMING UNITS are supplied from stock coils of wire. Operating sequence is: (1) Feed rolls direct 0.043 in. dia wire through top guide into guide bushing in ring forming unit. (2) When correct length, 2.30 in., has been exposed, impulse cam contacts rear of cutter. Cutter point hits wire flush with top guide. The wire is broken by impact. If not properly hardened, the wire will bend without breaking. If this happens, interlocks stop the machine. An automatic check of wire quality is thus provided. (3) After it is cut, the wire drops into small slot in the inserting mandrel. (4) With the cutoff wire in the bending fixture and in the slot in the inserting mandrel, the bending fixture is swung through 90 deg about an axis A at the end of the slot to the dotted position. This forms the hook, which stays in the slot in the mandrel. (5) Then the mandrel turns, as shown by the arrow, and draws the wire out of the bending fixture and into a half groove on the end of the mandrel. The formed wire is retained by the sleeve surrounding the mandrel. When the ring has been formed, the bending fixture returns to its starting position to receive the next piece of wire. (6) The inside of the sleeve is a few thousandths of an inch smaller in diameter than the piston pin hole and the end is beveled to fit closely against the piston side. The mandrel then moves sidewise and slides the locking ring into the piston boss until it expands into the groove in the piston. (7) Sleeve and the mandrel then retract together to receive the next piece of wire. A right and a left-hand unit operate simultaneously to form and insert retaining rings at both ends of the wrist pin. Ring forming units are driven by roller chains. The vee block which supports the wrist pin and the locating pin are mounted on hinged arms which drop below the piston to clear the forming units. In the actual installation, a canvas conveyor belt receives the completed assemblies as they fall from the machine.

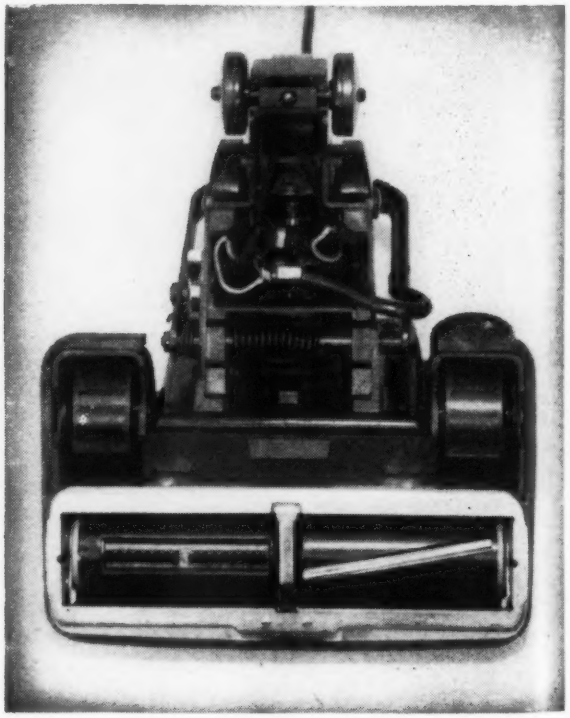
Molded Plastic Parts Simplify Vacuum Cleaner

Motor components of the new Hoover Junior vacuum cleaner are mounted in molded plastic parts instead of conventional metal housing to reduce weight and improve electrical insulation. The 14 lb machine cleans a 10 in wide path. It was developed to meet the demand for a more compact upright-type machine to fit the limited storage space available in modern apartments and small homes. Made by the Hoover Co., Canton, Ohio, it was styled by Henry Dreyfuss.

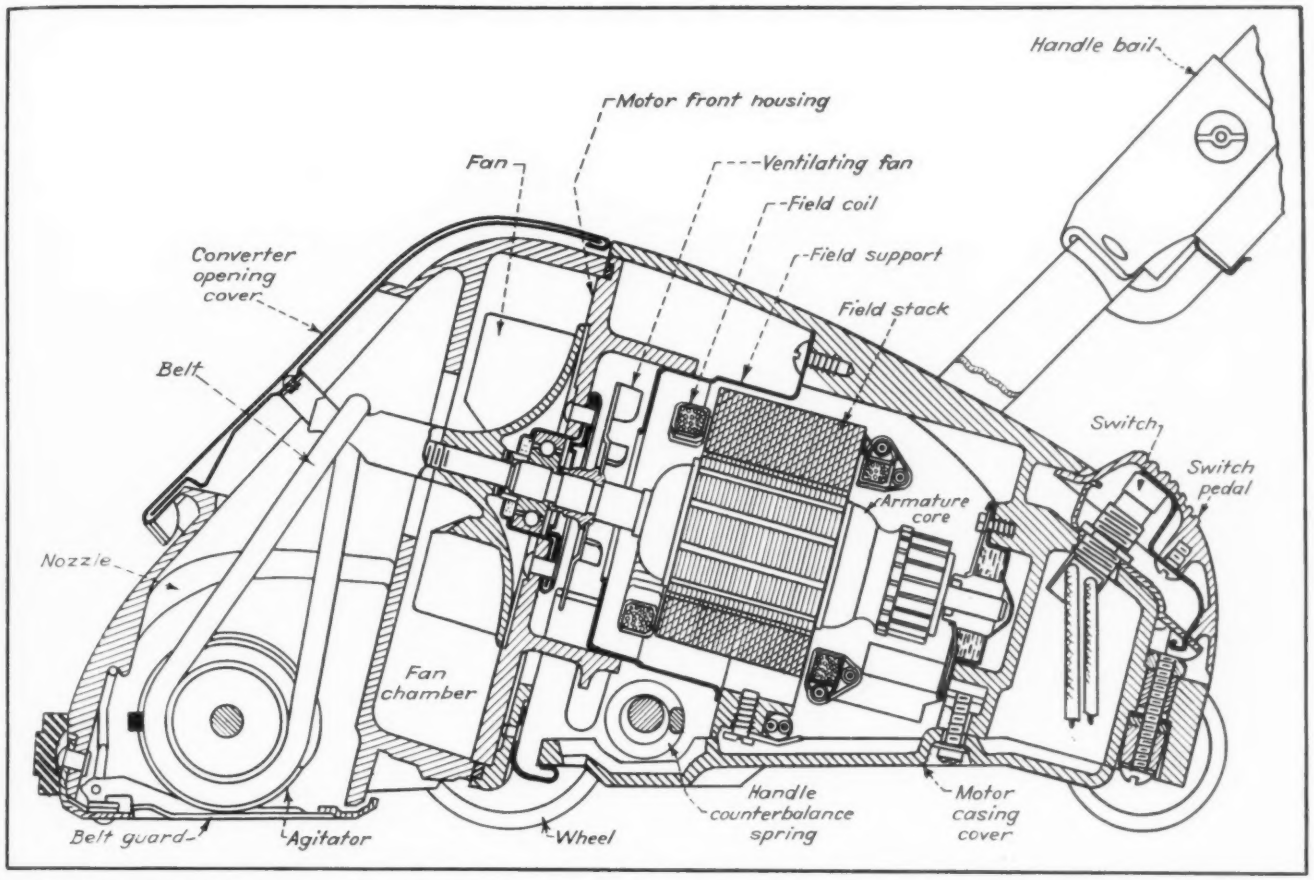
ATTACHMENTS are connected by removing front cover and inserting converter unit which connects to blower inlet. Lock-seam tubing proved cheaper than seamless for the handle.



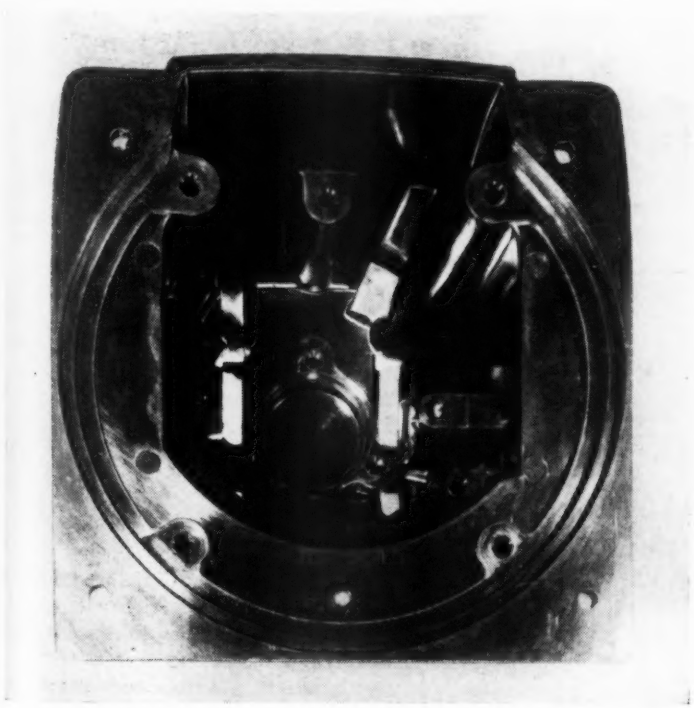
DIE CAST ALUMINUM fan chamber also houses beater assembly and front wheels. It required intricate coring to form passage to bag connection in upper center.



BOTTOM VIEW with cover removed shows motor wiring. Molded bottom cover is attached by one screw and retaining clip.

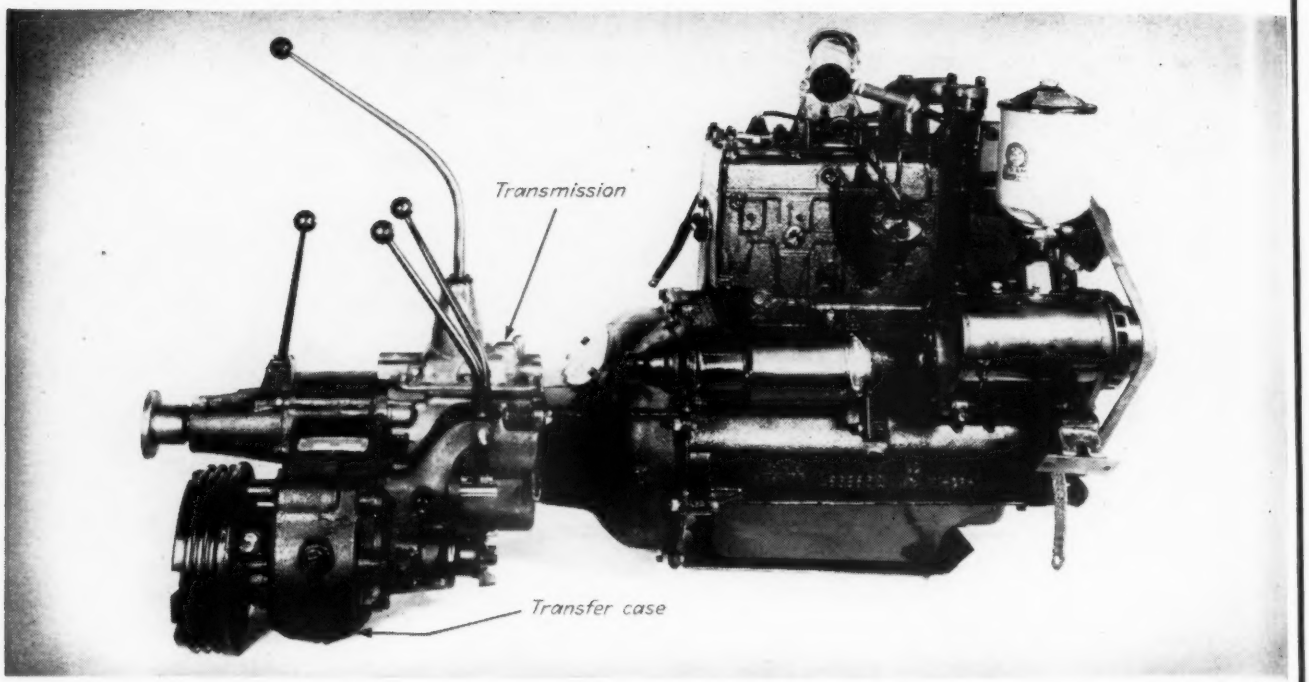


MOTOR is cooled by separate fan to protect bearings and commutator. Switch is located in rear of motor housing to simplify assembly and reduce hazard associated with wiring in handle. Cord is attached mechanically to handle to avoid interference with use of cleaner. Motor develops $\frac{1}{8}$ hp at 11,700 rpm.



MOTOR HOUSING and front and bottom covers are molded from semi-impact, flock filled, phenol formaldehyde compound. Motor components are mounted directly in the housing.

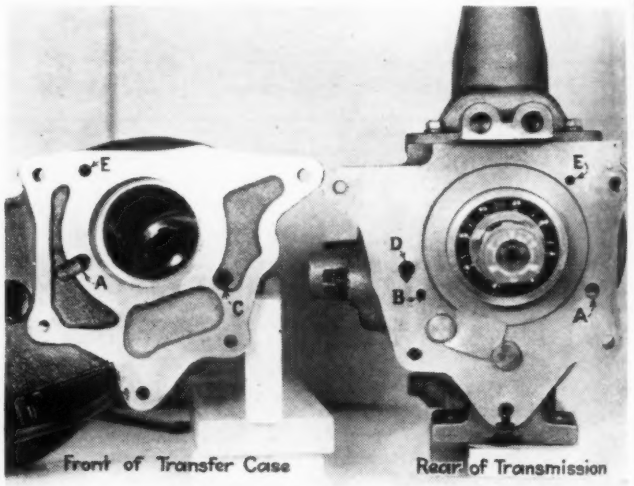
FRONT COVER (above) has molded slots for easy removal. Arrows (below) point to slots when cover is in place in housing.

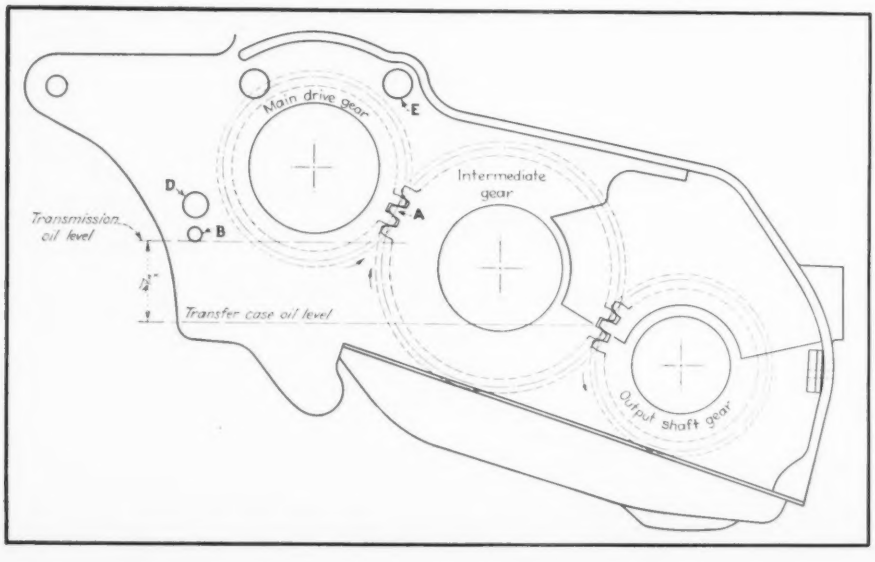


TRANSFER CASE of Willys-Overland "Jeep" contains low range gear set and connections to axle drive shafts.

Transmission Gear Lubricating System

Oil is pumped from transfer case to transmission by the drive gears and flows back by gravity in the "Jeep." The oil levels in the two housings differ by $1\frac{3}{4}$ in. Formerly each housing was filled and checked separately and an oil seal was required between the two units.



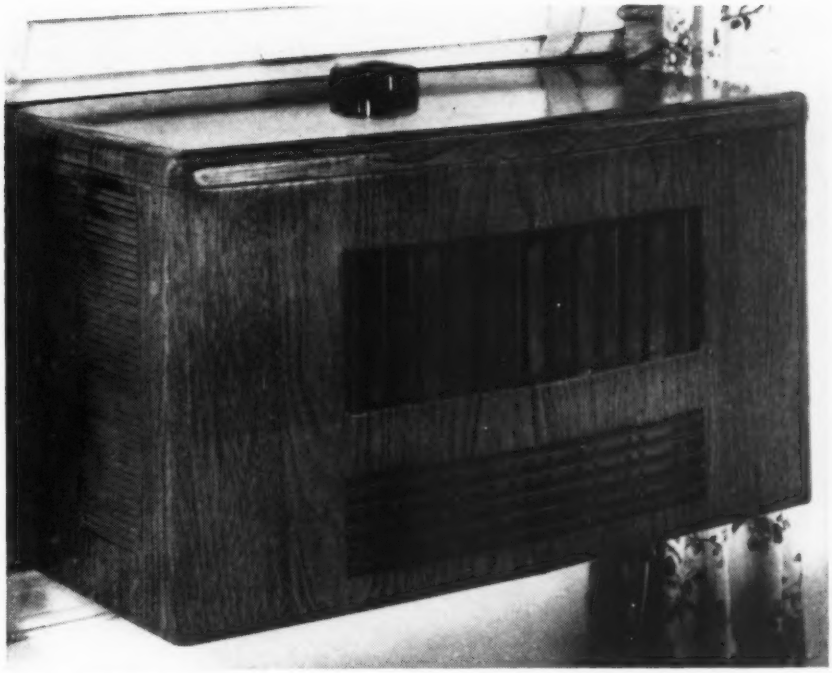


MATING FACES of transfer case and transmission. Oil is lifted through passages *A* from transfer case to transmission. Hole *B* allows oil to flow back to transfer case at the same rate. Hole *D* is an overflow. Hole *C* is oil return to transfer case. Hole *E* vents transmission.

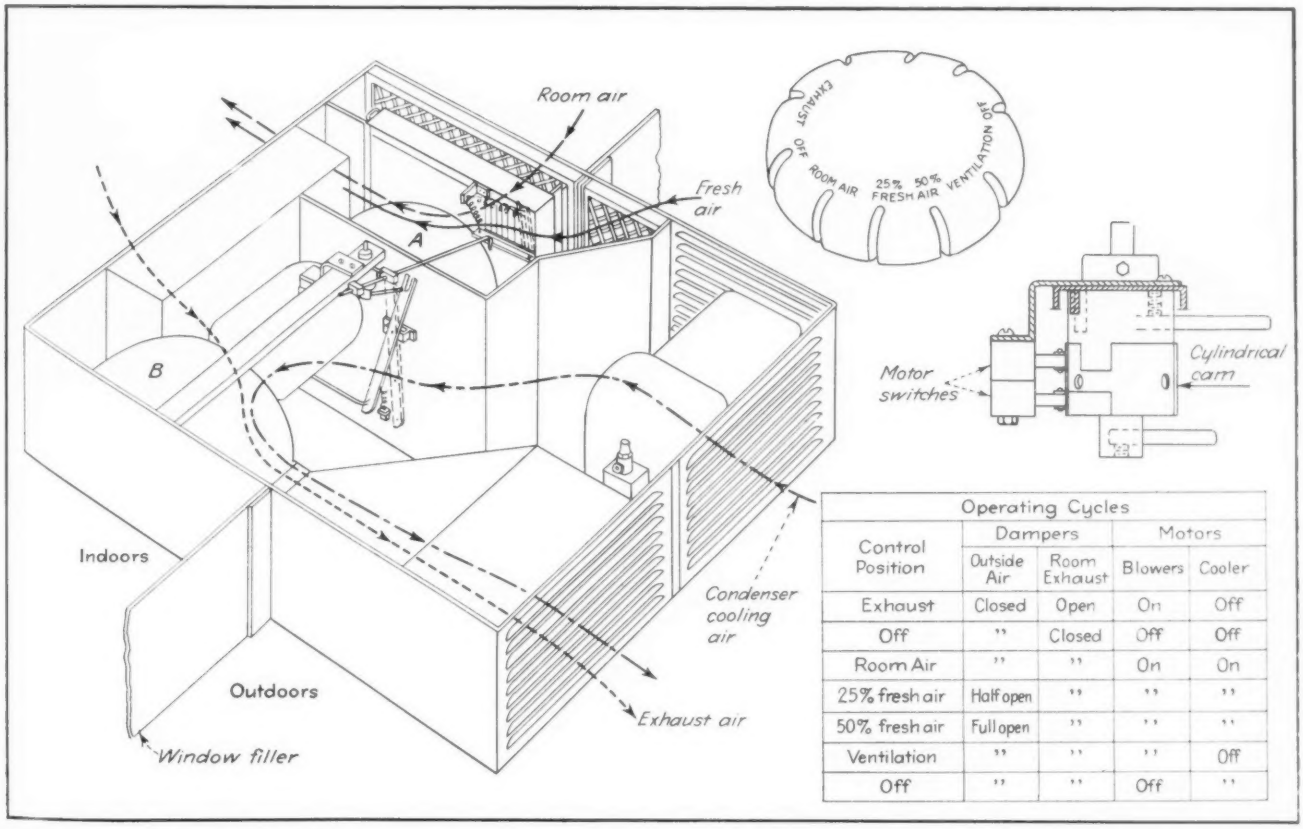
REAR VIEW of transfer case and transmission. Action of main and intermediate gears pumps oil at approximately 20 oz per min into transmission. Under all road conditions oil level is maintained within 15 percent of normal. Venting transmission through transfer case reduces likelihood of oil leakage from transmission.

Simplified Air Conditioner Control

Window-type air conditioner units made by U. S. Air Conditioning Corp., Minneapolis, Minn. can function as ventilators, coolers or exhausters. A major design problem was simple and foolproof coordination of compressor, blower, and damper controls. Charles Davies Associates, New York, N. Y. developed a single-knob control which also compensates automatically for dimensional variations resulting from manufacturing tolerances. Typical ½ hp unit weighs 250 lb and can remove 6000 BTU per hr.

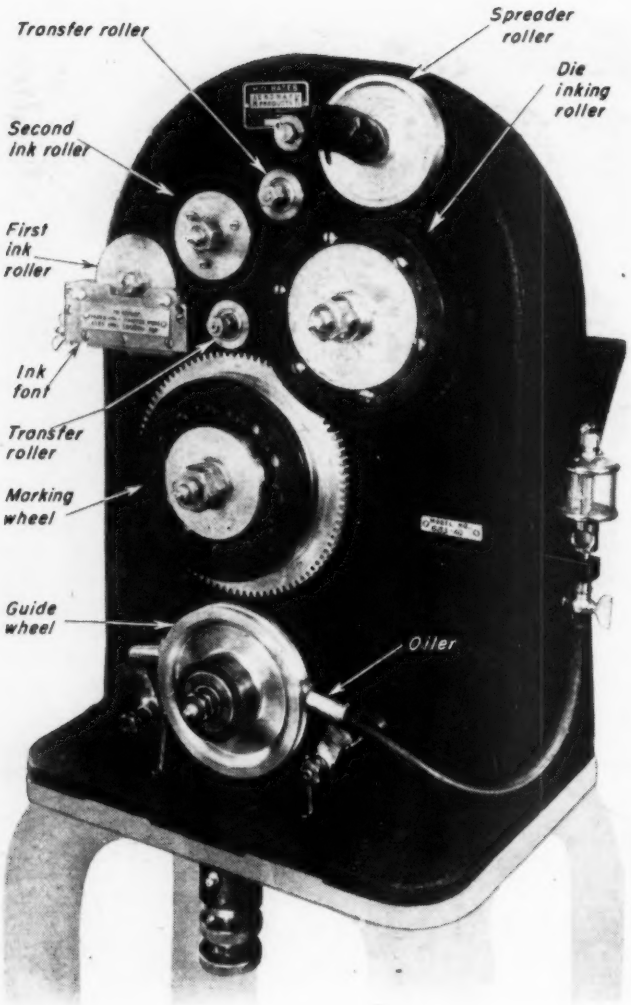


AIR CONDITIONER mounts in window opening. Control knob is on top surface. Upper front opening discharges air into room. Lower, is for exhaust. Louvres on left side are intake for room air to be recirculated.



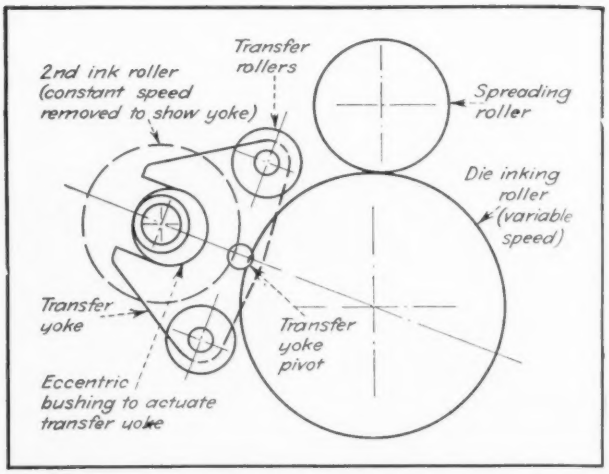
Operating Cycles				
Control Position	Dampers		Motors	
	Outside Air	Room Exhaust	Blowers	Cooler
Exhaust	Closed	Open	On	Off
Off	"	Closed	Off	Off
Room Air	"	"	On	On
25% fresh air	Half open	"	"	"
50% fresh air	Full open	"	"	"
Ventilation	"	"	"	Off
Off	"	"	Off	"

LOST-MOTION-LINKAGES actuated by control knob moves spring toggle mechanisms over center. Springs then move dampers to desired positions. This arrangement does not require adjustment to compensate for dimensional variations between assemblies. Fresh air damper toggle is nearly on dead center in "25 percent Fresh Air" position to help hold correct position. Blower *A* supplies incoming air to room. Blower *B* supplies cooling air for refrigerant condenser. When control is in "exhaust" position, *B* exhausts room air through exhaust damper. Raised portions of cylindrical cam operate switches. Design was planned to avoid special dies.

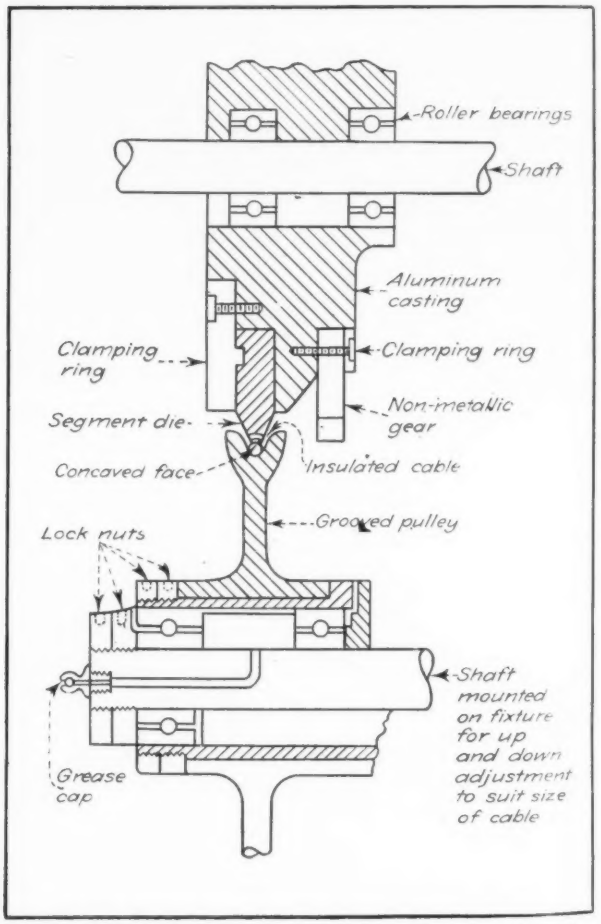


High Speed Marker

Variable winding speeds for electrical wire up to 1,000 ft. per min present unusual problems in applying uniform ink markings. The No. 683 Acromark machine uses an intermittent mechanism to charge ink transfer rolls by contact with a constant-speed inking system and discharge them by subsequent contact with the varying-speed die-inking roller. The marking wheel uses segmental dies arranged for quick change-over. Wire passes between groove in lower pulley and segmental dies (not shown) mounted in marking wheel. Marking wheel is driven by contact of dies with wire. Die-inking roller is geared to run in synchronism with marking wheel. Upper aluminum roller distributes ink over inking roller. Two movable transfer rolls alternately contact constant-speed inking unit at left and die-inking roller.


2nd ink roller (constant speed removed to show yoke)
Transfer rollers
Spreading roller
Die inking roller (variable speed)
Transfer yoke
Transfer yoke pivot
Eccentric bushing to actuate transfer yoke

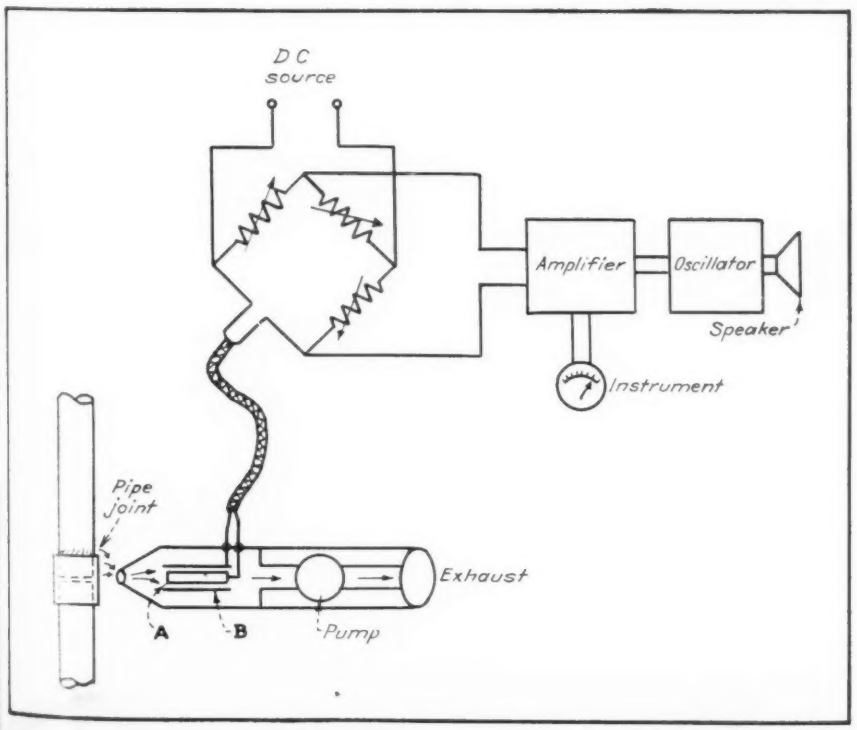
INTERMITTENT MECHANISM moves transfer rollers back and forth. This permits inking unit to run at the constant speed necessary for uniform ink supply while speed of marking unit varies between nearly zero and 1,000 ft. per min as wire builds up on winder core.



SEGMENTAL DIES are retained by ridge on clamping ring which engages groove in die. When cap screws have been loosened, clamping ring can be removed for die change by rotating a few degrees so that cap screw heads pass through large ends of keyhole slots. Interchangeable guide wheels are grooved to hold different sizes of wire. Spring mounting for guide wheel shaft deflects to pass enlarged spots or kinks. Oiler lubricates wheel groove to keep wire from sticking.

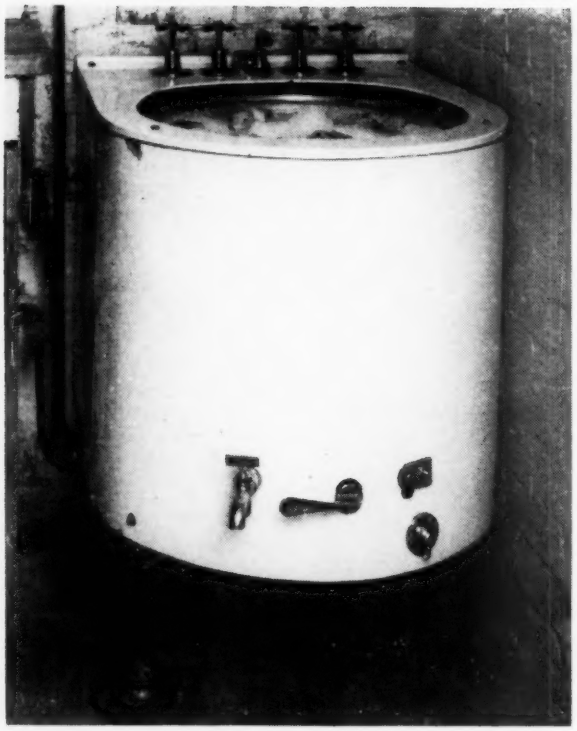
Sensitive Halogen Leak Detector

Changes in impedance of an air gap with changing atmosphere permit detection of halogen compound leaks, equivalent to as little as 0.01 oz per year of Freon, by the General Electric Type H Portable Leak Detector. Shop or field inspection of joints can be made at a rate of approximately ½ in. per sec. The unit was developed for testing mechanical refrigerating machinery but can be used on other systems by adding a volatile halogen compound to act as a tracer.

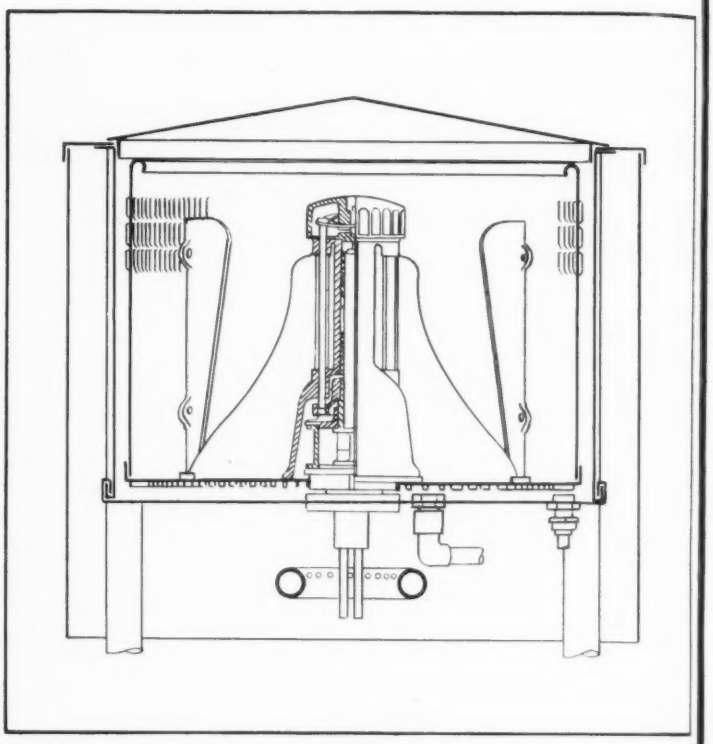


LEAK DETECTOR consists of 3 lb pistol-shaped detector unit and 15 lb control unit with carrying strap. Detector unit includes plastic-tipped sampling nozzle, sensitive element, d-c motor driven blower, and selenium rectifier for motor current. Control unit contains detector circuits and controls, indicating milliammeter, and power supply for 115 v 60 cycle input. Provision is made for earphones or speaker when aural indication is desired.

CIRCUIT DIAGRAM shows filament-heated central electrode *A* surrounded by platinum cylinder *B*. Sensitive element forms one arm of bridge circuit. Other arms of bridge are variable to adjust sensitivity and balance the circuit. When blower draws halogen vapors over detector its impedance decreases unbalancing bridge. Resulting voltage is amplified to give visual signal on meter and, if desired, aural signal from speaker.



CAPACITY of water driven Australian washing machine is 10 to 15 lbs dry weight of clothes depending upon whether they are boiled or not.

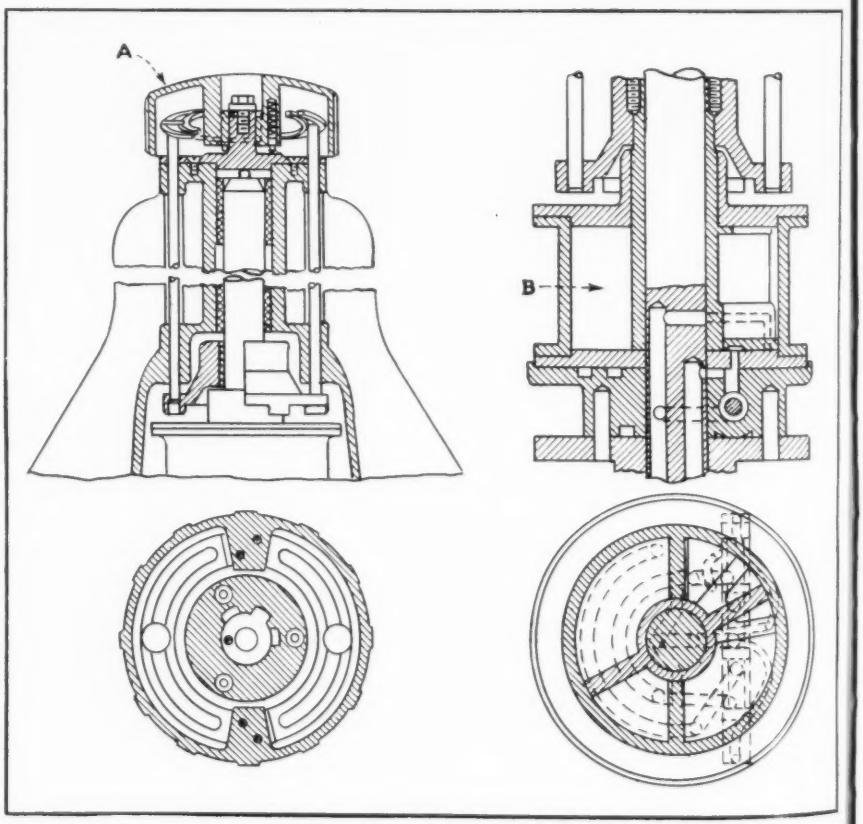


VANE MOTOR and pin clutch for oscillating drive are contained in central column. High speed turbine nozzle on bottom directs jet against blading on basket.

McGRAW-HILL World News

Water Power Washing Machine

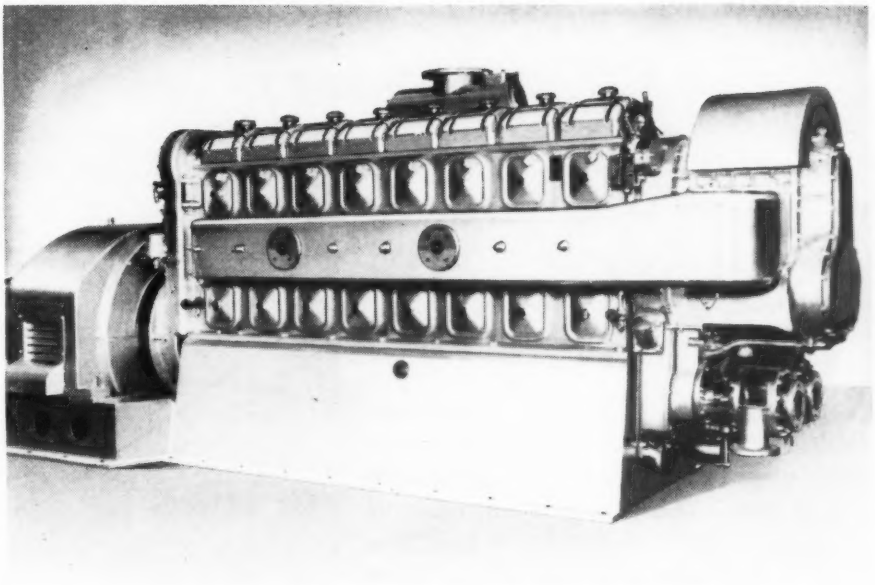
An oscillating vane water motor is used during the "wash" and "rinse" phases of the O.G. domestic clothes washer made in Australia. For drying, the vane motor is uncoupled by a pin clutch. A water jet is then directed against turbine blades on the bottom of the basket to spin-dry the clothes. The machine has an integral gas burner for heating the water or boiling the clothes if desired.



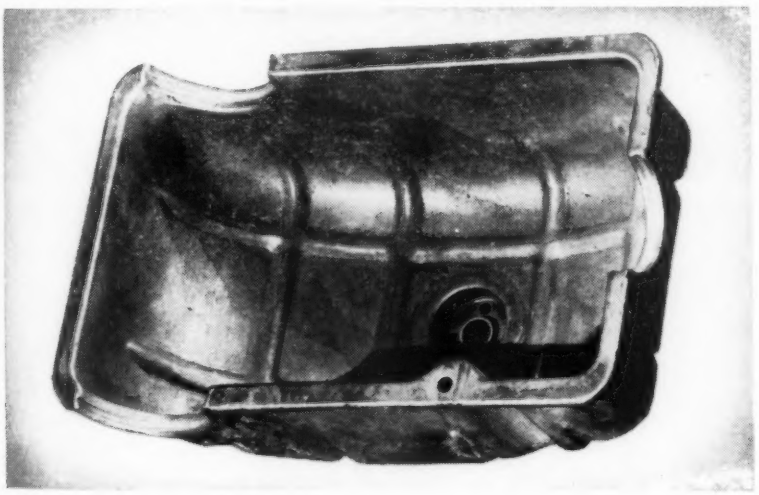
(A) ROTATION of control cap actuates plate cams which move clutch pins. Pins engage holes in driving flange of water motor. Central shaft is stationary. (B) Vane water motor oscillates through an arc of approximately 60 deg. Spool valve reverses pressure and discharge connections to motor chambers.

Stamped Cover Reduces Breakage

Drawn steel cylinder covers were adopted for heavy duty diesel engines to reduce breakage during servicing. Former aluminum castings broke when dropped but the steel stampings are easily hammered back into shape if dented. Material is 16 gage SAE 1010 hot rolled deep drawing stock. Parts are made by the Geometric Stamping Company, Cleveland, Ohio.

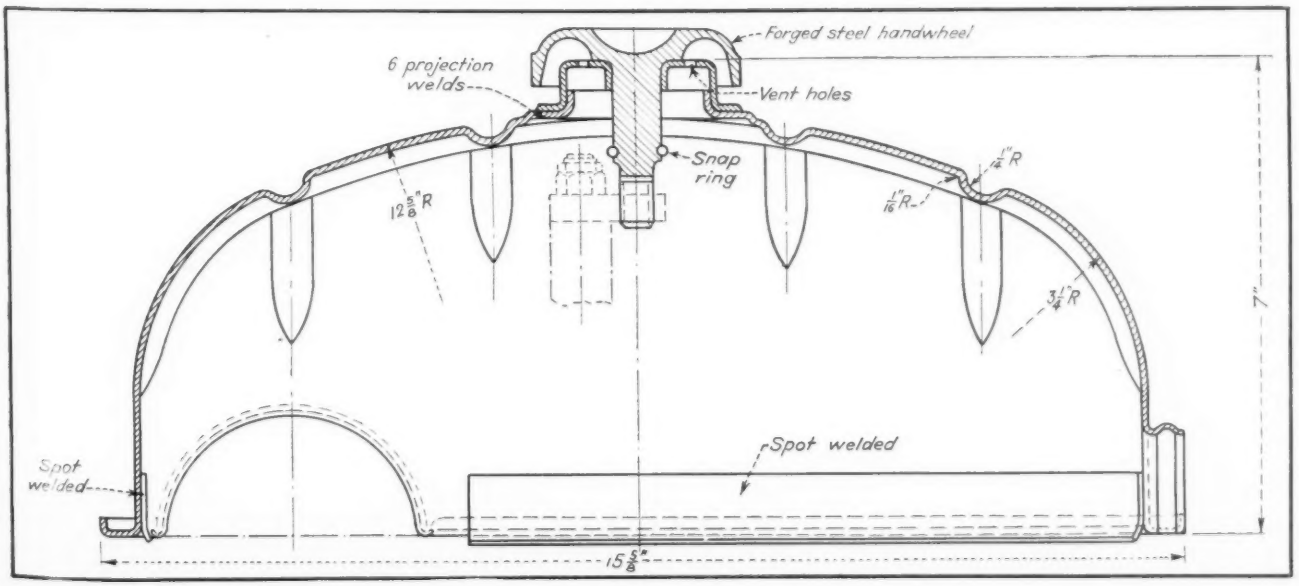


DIESEL ENGINE generator unit showing drawn steel covers over cylinder heads. Covers on side of engine were also changed to steel stampings.



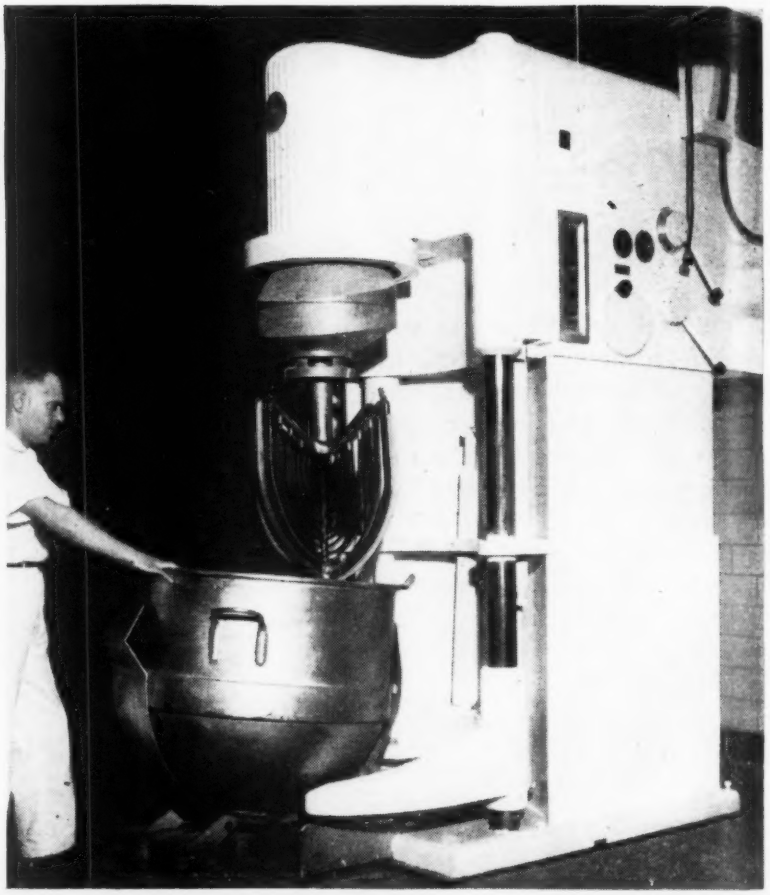
STAMPED COVER has stiffening ribs. Part is shown before spot welding gasket retainers around lower inside edges. Parts are soldered after welding to prevent oil leakage. The completed, 7 lb assembly is then electro-tin plated. Composition gaskets are cemented to sealing faces.

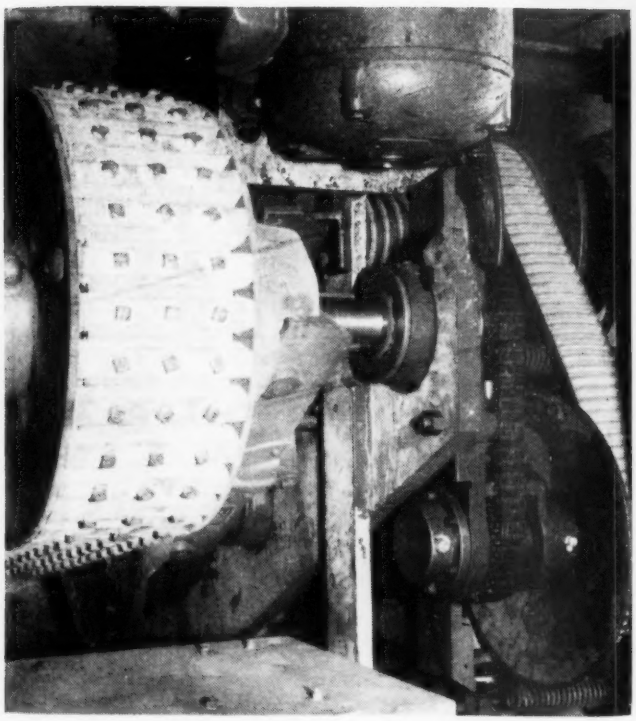
SECTIONAL VIEW of cover shows gasket retainers welded around bottom. Drawn 12 gage cup, welded to top, guides captive hand wheel which screws into bracket on engine. Hand wheel also forms dust shield for vent holes. Six stamped ribs minimize distortion of cover.



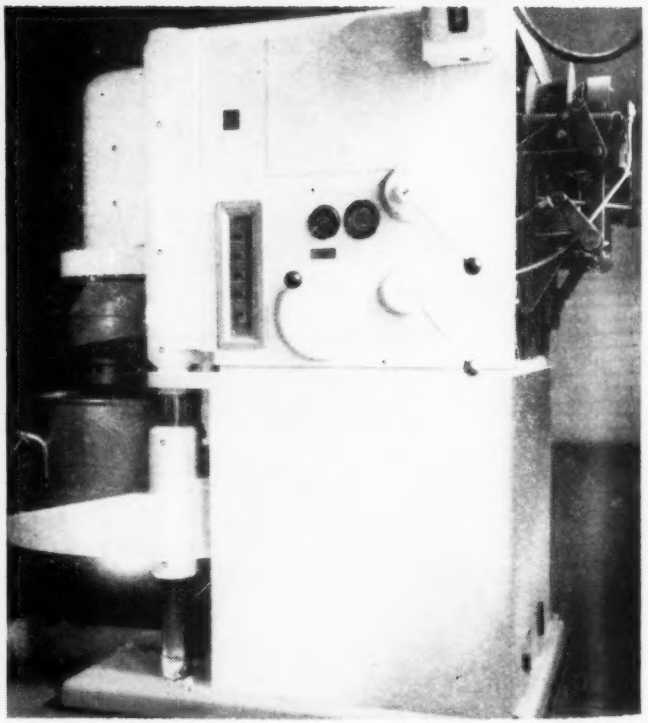
Variable Ratio and Speed for Dough Mixer

Two variable speed drive units permit independent selection of beater speed and ratio of beater speed to precession of beater assembly around bowl. This 340 qt mixer, made by American Machine and Foundry Co., New York, N. Y. is used for mixing abrasives and chemicals as well as bakery products. Mixer head is raised and saddle lowered to receive 85 gal bowl. Beater design and materials are selected to suit materials to be mixed. Vertical, notched bar locks upper frame and bowl together during mixing cycle.

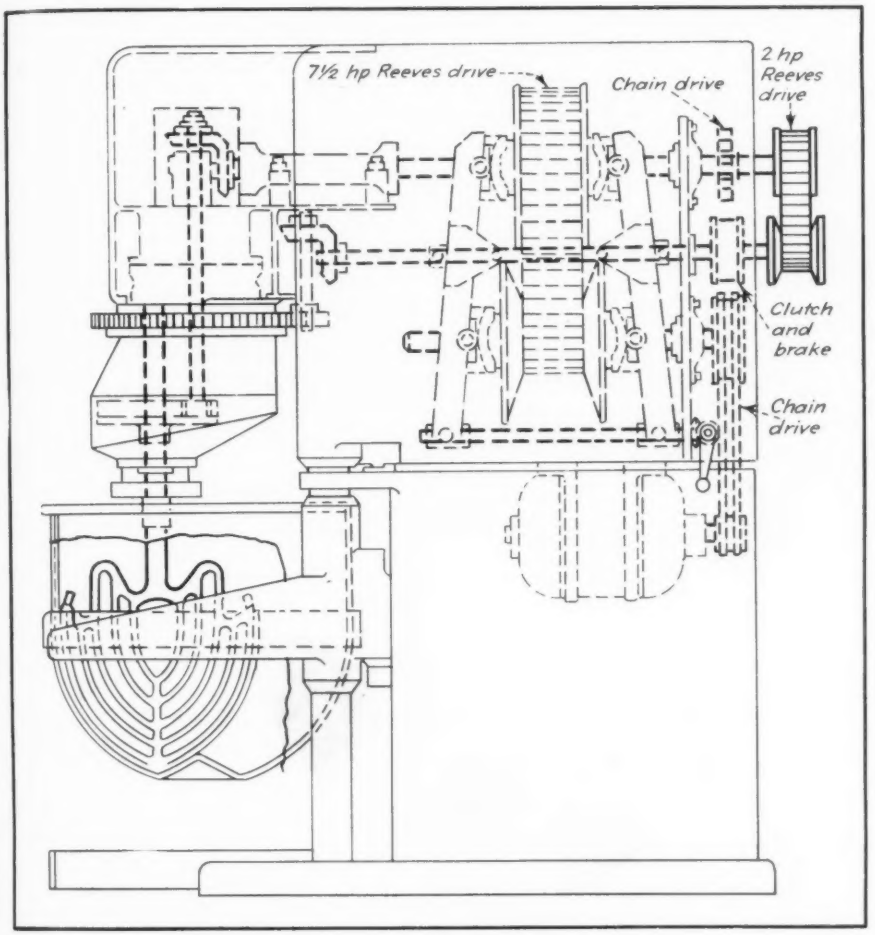


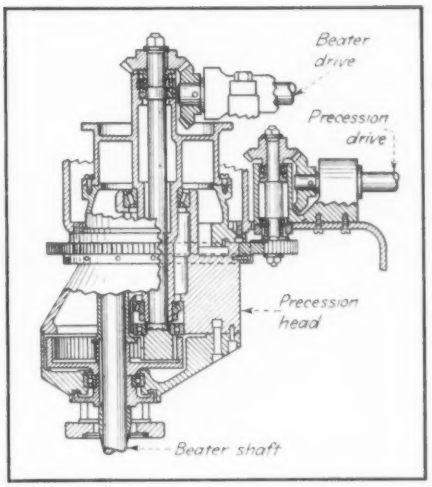


TOP VIEW shows 7½ hp variable speed drive in left center connected to beater-rotating shaft. Chain at right drives 2 hp variable speed unit. The latter drives beater-precession shaft through cone clutch (not shown). The motor in the upper part of the picture raises and lowers the mixing head and bowl-carrying saddle.



MIXING POSITION is obtained by lowering upper assembly and raising saddle and bowl. Movement is accomplished by 1 hp motor in upper assembly. Rear cover has been removed to show 2 hp variable speed unit for precession drive at upper right. Timer and tachometer are provided to promote uniformity in operation.



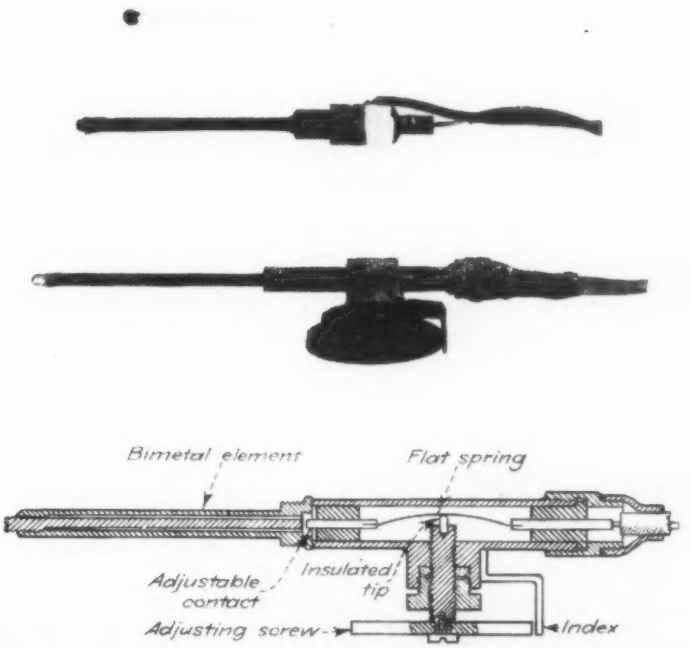


DRIVE SYSTEM receives power through motor, chain drive and 7½ hp variable-speed unit. Ratio of beater rotating speed to precession speed around bowl is important. Hence precession drive is taken from variable speed beater-rotating shaft rather than from constant speed input shafts. The beater head is rotated by a ring gear and pinion. A coaxial shaft and pinion engage an internal gear which is connected to the beater.

McGRAW HILL *World News*

Sensitive Adjustment for Thermostat

The adjustable contact of the Tem-Con thermostat is positioned by a toggle-like combination of a flat spring and a fine pitch screw. Transverse movement of the adjusting screw changes the deflection of the spring and causes axial movement of the adjustable contact. With this arrangement a large movement of the screw results in a small movement of the contact. The fixed contact is actuated by a bimetallic element 5/32 in. dia and 2 in. long. The manufacturer, P.A.M. Limited, Merrow, Guildford, Surrey, England say that this unit will respond to changes of 0.1 deg C.



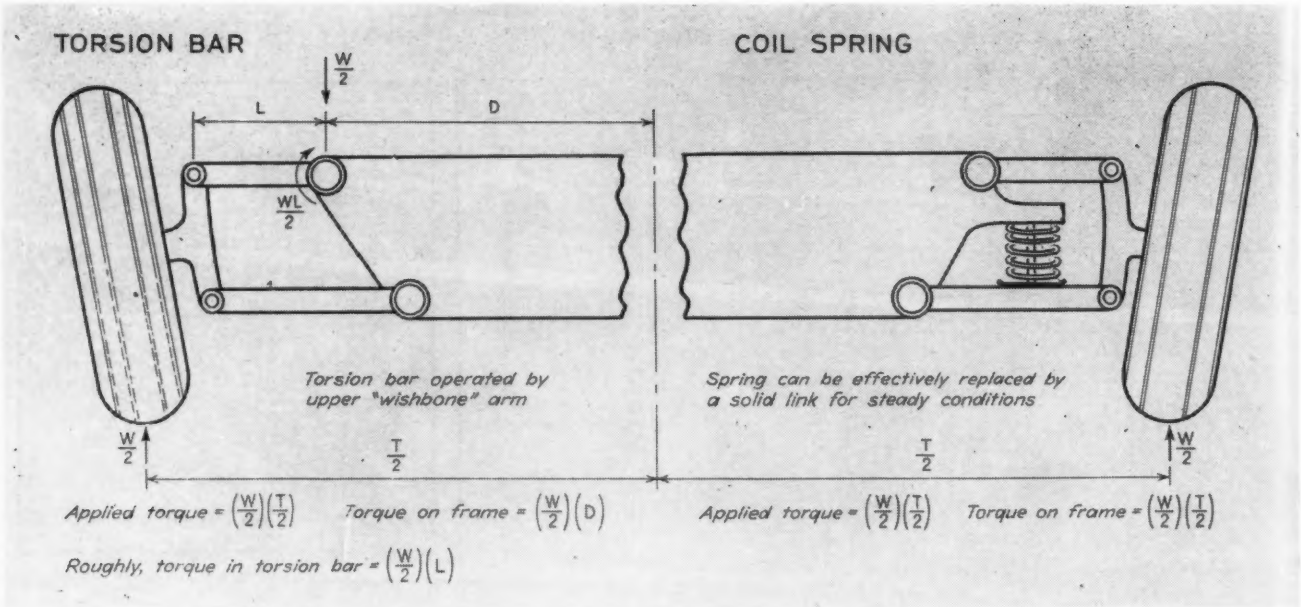


Fig. 1—Diagram showing how an automobile with torsion bar independent front suspension applies less torque to its structure under conditions of unequal front wheel loading than a similar car with a coil spring independent front suspension.

Applying Torsion Bar Springs

Methods of shortening spring length. Relationships between energy storage, stress level, diameter, lever length, and load.

DONALD BASTOW
Assistant
W. O. Bentley, Consulting Engineer

TORSION BARS can be used either to transmit the applied torque from the input point to some other relatively distant point, or to bring that torque back to somewhere near its point of application. An example of the first is the conventional automobile application for independent front suspension. The sketch, Fig. 1, shows how the torsion bar in conjunction with a wishbone suspension reduces the torque applied to the car frame or structure proper during conditions of unequal front wheel loading. Common causes of this condition are cornering and one wheel only striking a bump. Torque on the frame is reduced by the amount of the torque taken by the torsion bar itself. This should either permit the use of a frame that is torsionally more flexible and hence lighter, or alternatively confer the benefits of greater torsional rigidity in a given frame. An example of the second is the Vauxhall car application already referred to in Figs. 5 and 6, "How To Design Torsion Bar Springs", PE, September, 1948.

The torsion bar itself and its attendant parts must be made to suit the circumstances of its installation. This will become clear from study of the graphs, Figs. 2 through 8, showing how torsion bar weight and volume vary with the maximum stress employed, how torsion bar diameter varies with stress, and with load, and how torsion bar length varies with lever length. These all illustrate how one limitation in an installation brings in its train further restrictions. A restricted torsion bar length, for instance, means a long lever or a large diameter torsion bar, or a number of torsion bars of smaller torque capacity in parallel (as in the German KdF car) and a high anchorage torque. If the lever length is restricted, the torsion bar must be long, or be "doubled back," or consist of a number of bars in parallel. The design process consists in finding the best compromise between the more or less irreconcilable requirements. Here again this may be best expressed by taking an example.

Let the normal load be 700 lb and the desired rate be 100 lb per in. From the formulas given, the rate for a solid round bar is

$$\text{Spring rate} = \frac{\pi d^4 G}{32 l L^2} \text{ lb per in.}$$

and the stress in the material is

$$f_s = \frac{16\,WL}{\pi d^3} \text{ psi}$$

From the first of these

$$3{,}200\,lL^2 = \pi d^4 G$$

and from the second (assuming again a normal stress of 70,000 psi)

$$70{,}000\,\pi d^3 = 16 \times 700 \times L$$

If, from a rough scheme layout, it looks as though L should be 12 in., then

$$d^3 = \frac{16 \times 700 \times 12}{70{,}000 \times \pi} = 0.6105$$

and therefore

$$d = 0.848 \text{ in.}$$

From this it follows that

$$l = \frac{\pi \times 0.518 \times 11.5 \times 10^6}{3{,}200 \times 144} = 40.55 \text{ in.}$$

If there is insufficient room available to accommodate this length of torsion bar, L must be increased above 12 inches. An increase of 1 in. to 13 in., for instance, reduces the required length to 38.6 in., a decrease of almost 2 inches. An alternative means of reducing the length would be to increase the stress; still further alternatives are to use a number of bars in parallel or to "double back." Considering the former of these first, it is obvious that the sum of the d^3 figures of the nest of bars must equal d^3 of the single bar. The result is that for any number, N, of torsion bars in parallel, the length of torsion bars required, as a percentage of

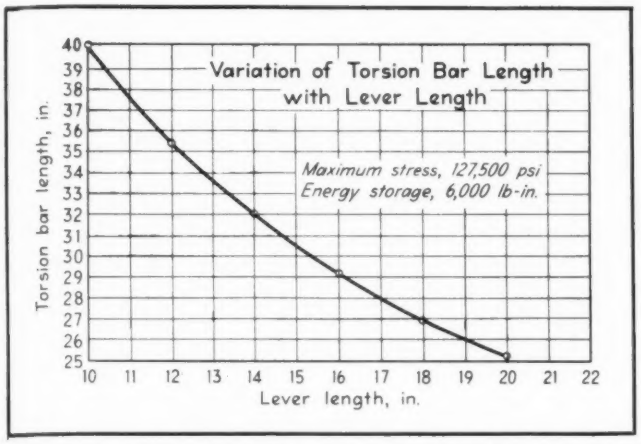

Variation of Torsion Bar Length with Lever Length
Maximum stress, 127,500 psi
Energy storage, 6,000 lb-in.
Torsion bar length, in.
Lever length, in.

Fig. 2—Graph showing how torsion bar length varies with change in operating lever length, for a fixed maximum stress.

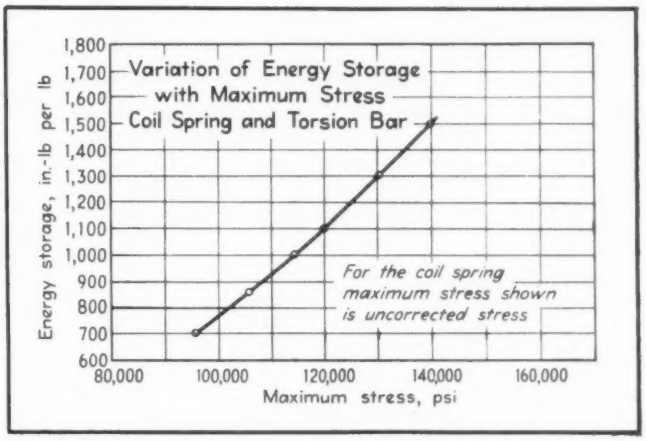

Variation of Energy Storage with Maximum Stress
Coil Spring and Torsion Bar
For the coil spring maximum stress shown is uncorrected stress
Energy storage, in.-lb per lb
Maximum stress, psi

Fig. 3—Graph showing how energy storage per lb in a coil spring or in a torsion bar varies with the maximum stress.

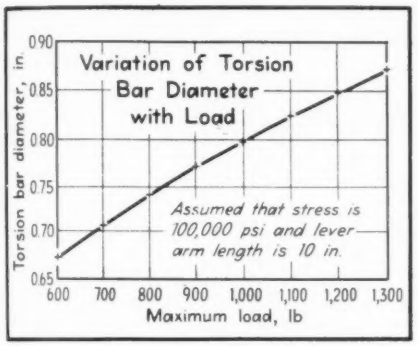

Variation of Torsion Bar Diameter with Load
Assumed that stress is 100,000 psi and lever arm length is 10 in.
Torsion bar diameter, in.
Maximum load, lb

Fig. 4—Graph showing how torsion bar diameter varies with load, for an assumed lever length of 10 in. and a 100,000 psi maximum stress in the spring material.

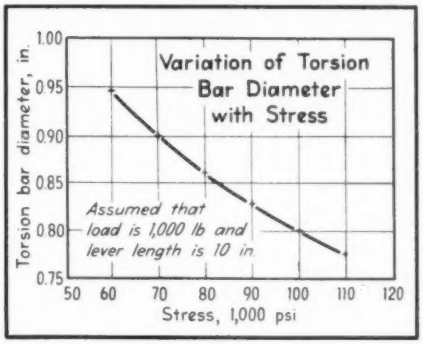

Variation of Torsion Bar Diameter with Stress
Assumed that load is 1,000 lb and lever length is 10 in.
Torsion bar diameter, in.
Stress, 1,000 psi

Fig. 5—Graph showing how torsion bar diameter varies with the maximum stress, for an assumed load of 1,000 lb and a 10 in. operating lever length of the spring.

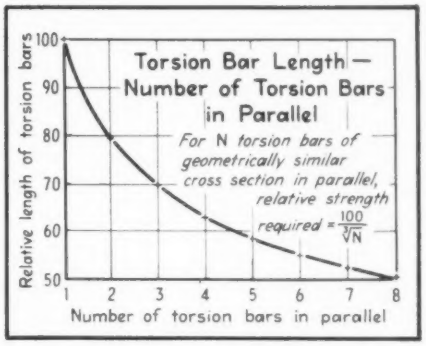

Torsion Bar Length — Number of Torsion Bars in Parallel
For N torsion bars of geometrically similar cross section in parallel, relative strength required = 100/∛N
Relative length of torsion bars
Number of torsion bars in parallel

Fig. 6—Graph showing how torsion bar length varies with the number of torsion bars of similar cross section used in parallel to shorten space requirement.

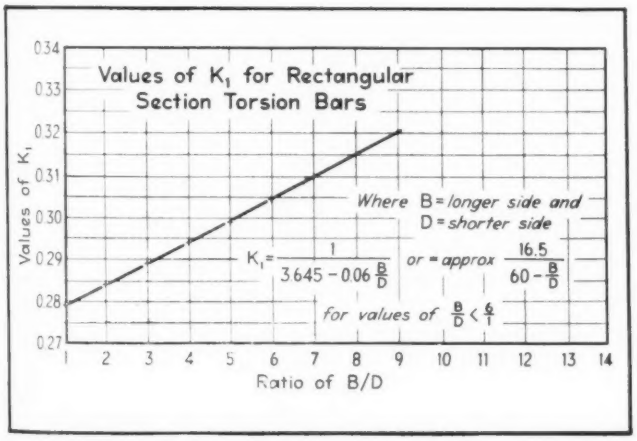

Values of K1 for Rectangular Section Torsion Bars
Where B = longer side and D = shorter side
K1 = 1 / (3.645 − 0.06 B/D) or = approx 16.5 / (60 − B/D)
for values of B/D < 6/1
Values of K1
Ratio of B/D

Fig. 7—Values of K_1 required for use in formulas for rectangular section torsion bars given in PE, September 1948, p 112-113.

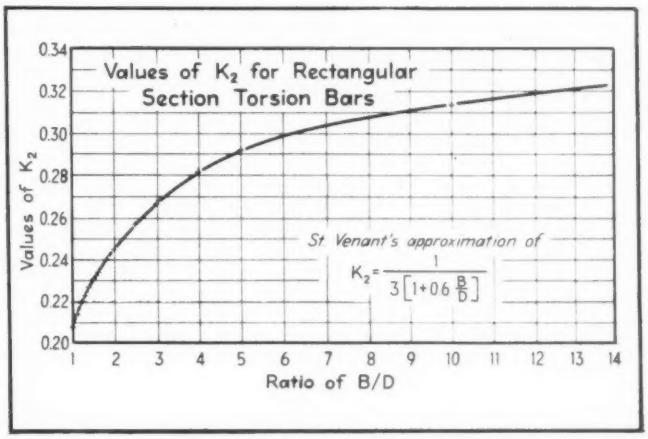
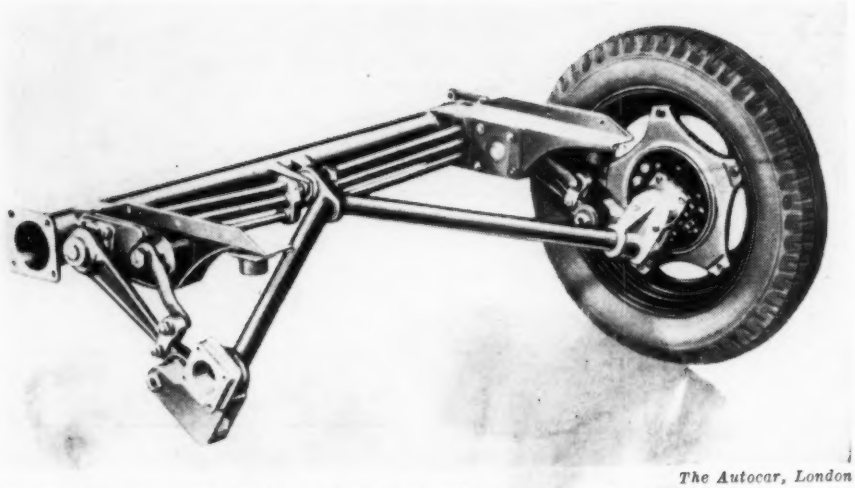


Fig. 8—Values of K_2 required use in formulas for rectangular section torsion bars. Based on St. Venant's approximation.

that required for one torsion bar, is $100/\sqrt[3]{N}$. This is shown in the graph, Fig. 6, of torsion bar percentage lengths against the number of torsion bars in parallel. "Doubling back" thus may reduce the benefits derived from the use of torsion bars as previously mentioned.

Apart from physical limitations of this kind, there are other things to be borne in mind in arranging the torsion bar suspension layout. One of these is the desirability of avoiding friction in bearings in the suspension layout and the need for lubricating those bearings. Both of these conditions lead to the use of rubber bearings in the suspension linkage and torsion bar mounting. It must be remembered that such rubber bearings have a torsional rate that will have the effect of increasing the spring rate at the wheel, without having increased the load carrying capacity. Another problem in motor vehicle applications is that road noises are liable to travel from the road wheel up the torsion bar to some point at which they may be heard in the vehicle. Much of this can be avoided by interposition of rubber at the appropriate points, either in the connections, mounting or anchorage.

An unusual torsion bar suspension is shown in Fig. 9. The two independent torsion bar assemblies are



The Autocar, London

Fig 9—Rear suspension of the Dyna-Panhard. Central torsion bars are splined at inner end to the floating T-heads. Upper and lower torsion bars are splined to T-heads and outer frame. Tubular Vee hub support is pivot mounted on the frame cross piece.

not joined together at the center of the axle, as might appear at first glance.

Each rear wheel has its own separate torsion bar assembly, comprising a central bar, 21 mm dia running close behind and parallel to the rear tubular frame member, transversely across the frame. It is roughly 15.5 in. long. This central bar on each side is splined to the radius arms that support each separate wheel hub. The inner end of each central torsion bar is splined to a steel T-head block, which is a floating member, and is itself connected to the car frame by a pair of torsion bars, each 18 mm in diameter. These parts are splined to the main frame near the sides of the car.

The springing action of the three bars in combination has been likened to the effect of "twisting the central prong of a pitchfork while holding the tips of the two outer prongs firmly."

The hubs are supported on a tubular Vee-shaped cross member, located by a central pivot. Further fore-and-aft location is provided by the trailing radius arms. Net result says *The Motor,* is "to give greater flexibility at the rear combined with an almost fantastic stiffness in roll."

One of the weaknesses of the torsion bar in the past has been the conditions at its ends. Modern practice is to make them larger with 25-35 percent greater diameter as compared with the active part of the bar; to use either Vee serrations with an ample radius at their roots (so that the shot can penetrate there and protect the points at which stress concentrations exist); or else to use some form such as a hexagon, which has even less tendency to act as a stress raiser. In most automotive applications there will never be reversal of load at the ends, and there is no need to fear trouble from fretting.

New Plastic Resists High Heat and Corrosion

A NEW, STABLE high temperature thermoplastic will find many engineering applications according to recent disclosures by The M. W. Kellogg Co., Jersey City, N. J. The plastic is a polymer of trifluorochloroethylene, four-fifths of its weight consisting of the two halogens, flourine and chlorine. In its natural state it is colorless and transparent but it can be blended with solid fillers and coloring agents. Tradenamed "Kel-F", this material was developed to the commercial production stage by Kellogg scientists in consultation with Dr. W. T. Miller of Cornell University.

Polytrifluorochloroethylene is closely related to the family of organic compounds — the fluorocarbons — which reached practical development during the war. However, it differs from the fluorocarbons in that some of the fluorine is replaced by chlorine. Fluorocarbons are similar in structure to the hydrocarbons of petroleum, but differ from such molecules because all of the hydrogen—which is known to be a point of chemical attack—is replaced by fluorine. Because of this replacement, fluorocarbons are noted for chemical inertness.

The new polymer exhibits the high stability properties of the fluorocarbons combined with many commercially useful physical properties. For example, although extremely resistant to chemical action, it is easily worked in conventional equipment; it is strong and hard but not brittle; it can be used at high temperatures but is satisfactory at very low temperatures. Can be heat-treated similar to tempering of metals.

Stable properties at use temperatures are obtained by suitable heat-treatment at considerably higher temperatures. In this way the plastic can be made relatively soft and resilient, resembling a plasticized material, or harder and less yielding. No sacrifice of any of the stability characteristics of the plastic is made under heat-treatment. No plasticizers are used.

The material can be used for pump packings, valve seats and valve packings wherever corrosive gases and liquids are processed. Because it is transparent, it can be substituted for glass in gage glasses, flowmeters and other applications where glass might shatter or be chemically attacked. It can be laminated with wire or glass cloth where chemical inertness of a plastic with additional mechanical strength is desired. It deforms under load but returns to original shape when load is removed.

The plastic is a good electrical and heat insulator and will probably make good wire insulation. It is thought that such insulation for electrical equipment will permit operation at higher temperatures.

Polytrifluorochloroethylene has good fabricating characteristics and can be produced by conventional machinery designed to operate between 550 and 600 F. It can be compression, transfer or injection molded, or extruded. It has been formed in sheets, films, tubing, rod, shaped pieces or as insulation on wires. It can be punched or drilled easily and can be machined to close tolerances.

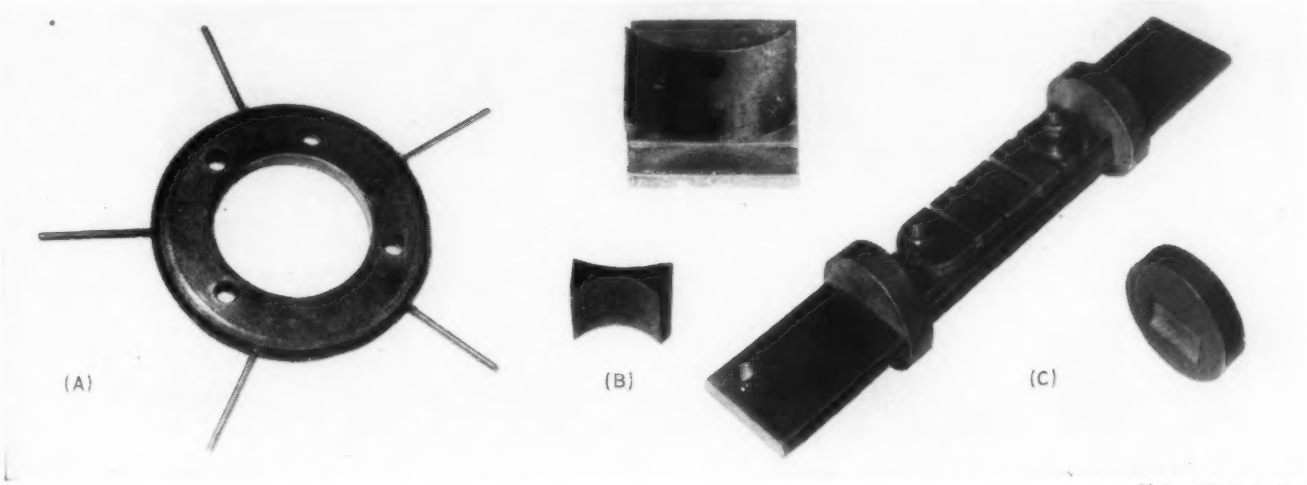
Table of Physical Properties

Specific gravity	2.1
Tensile strength at 86 F, psi.	9,400
Elongation at break 86 F, percent	127
Coefficient of linear expansion (−320 F to 167 F)	3.5×10^{-5}
Specific resistance, ohms	5×10^{17}
Dielectric strength short time volts/mil (0.006 in.)	2,500
Dielectric constant at 60 to 10^6 cycles	2.3 to 2.7
Water absorption, percent	0.00
Outdoor weathering	no detectable change
Temperature range, F	−320 to 390

Nonferrous Powdered Metal Parts

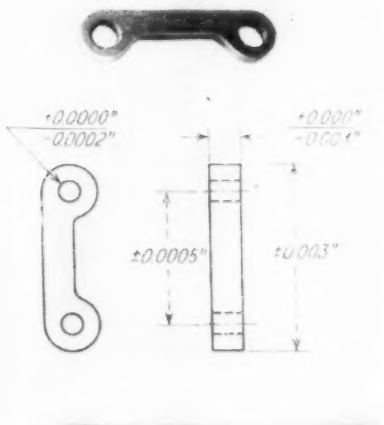
Typical machine elements pressed of copper, brass and nickel silver. Some cost less or are more accurate than competitive parts. Others have improved electrical characteristics.

D. C. BRADLEY
Market Development Division, The New Jersey Zinc Company



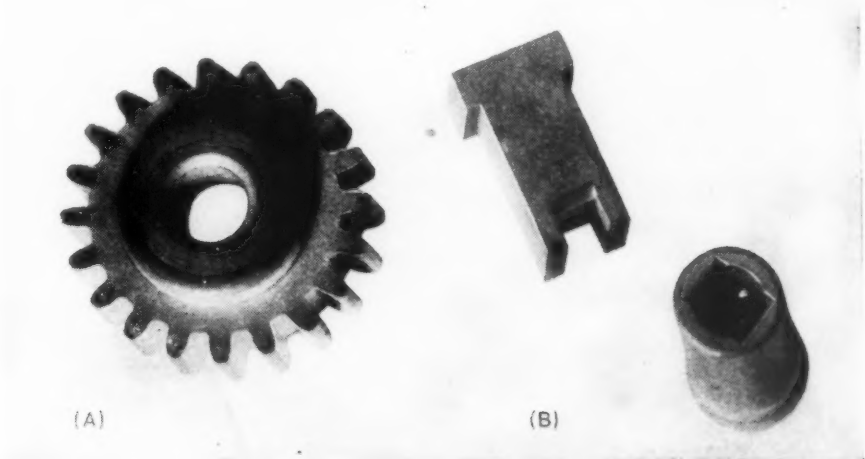
Sintered Metals, Inc.

(A) ALTHOUGH THE COST of this spark rotor was less when produced by die casting, pressing it of copper powder results in better electrical conductivity. Surface uniformity and appearance also are improved. The tungsten inserts are hand loaded in the dies, and the parts are sintered in special fiixtures. (B) These copper saddles for electrical parts must be produced with the equivalent of a lapped surface on the curved sections. Machining them from solid stock is impractical because of tool marks. When they are pressed and coined, the finish is satisfactory and the cost is about half that of machined parts. The production rate is high, the anode saddle on the right being made at a rate of 500 pieces per hour. (C) The fuse washer shown here was formerly stamped but now is made of powdered brass having a composition of 85 percent copper and 15 percent zinc. It is said to have a more uniform surface, better appearance, and greater dimensional accuracy. Costs are 20 percent less.



General Sintering Corporation

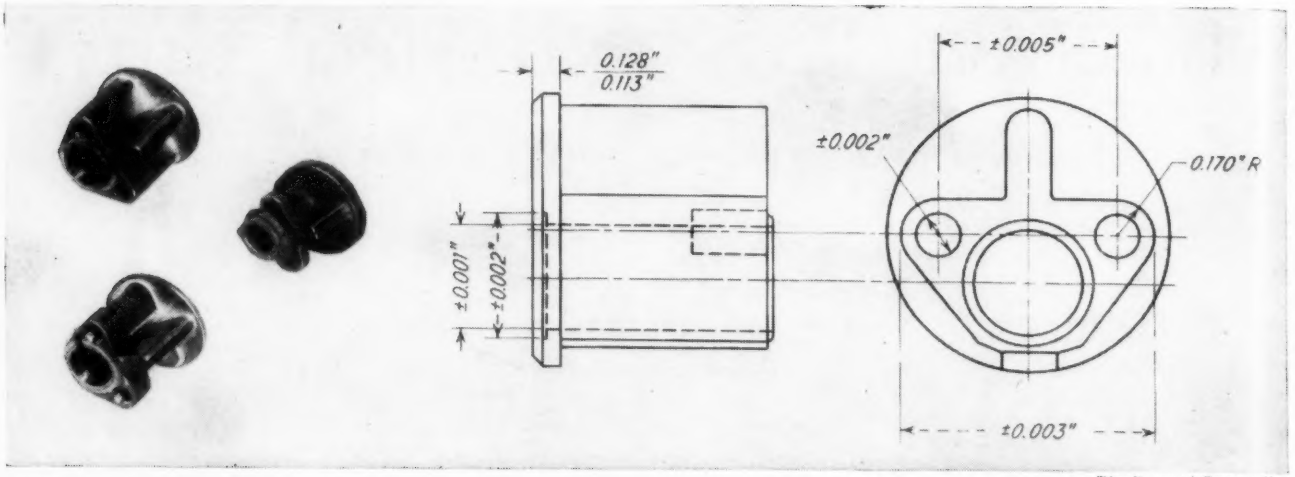
SINTERED POWDER parts often are impregnated with a lubricant, as is the link element, shown here, to decrease operating friction. Its porosity is held to approximately 30 percent to permit oil impregnation. Minimum shrinkage is obtained with a 75 percent brass—25 percent copper powder mixture. Note the close tolerance specified.



American Sintered Alloys, Inc.

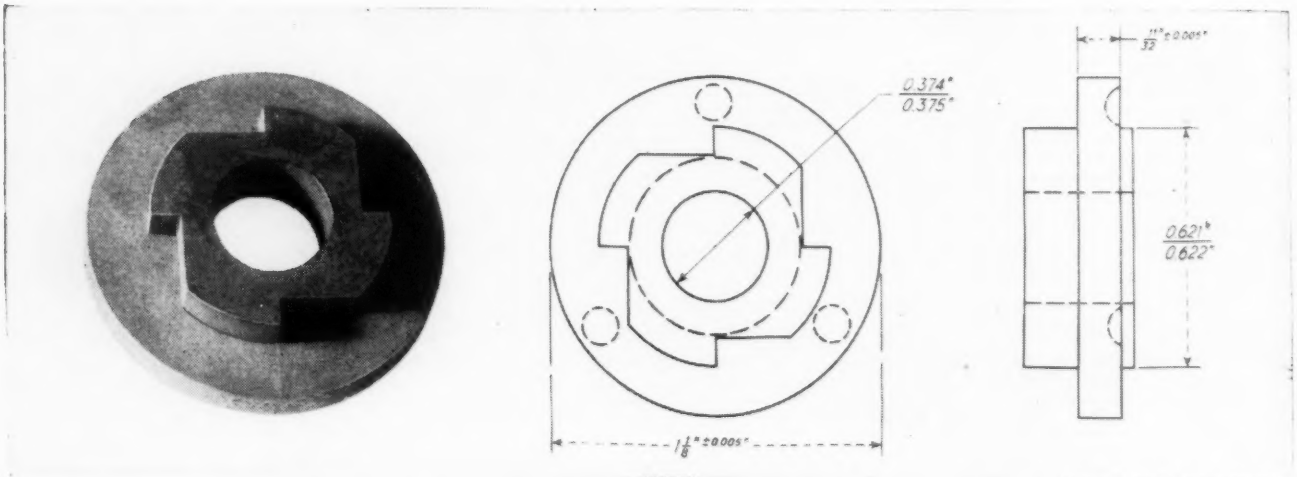
(A) This small, lightly loaded speed change spur gear is made of 85-15 brass powder. It is oil impregnated, wears slowly in service, and costs 35 percent less than a similar unit machined from bar stock. If necessary, key slots can be included in the mold so that subsequent machining operations are unnecessary. (B) The brass brush holder, right, is an insert for a molded plastic assembly. Its high surface porosity results in a better bond between the plastic and the metal than could otherwise be obtained. The advantage of making the rheostat contact, center, from powdered metal is that the electrical conductivity is high and the cost is 25 percent less than that of a machined or extruded part.

(*continued on next page*)



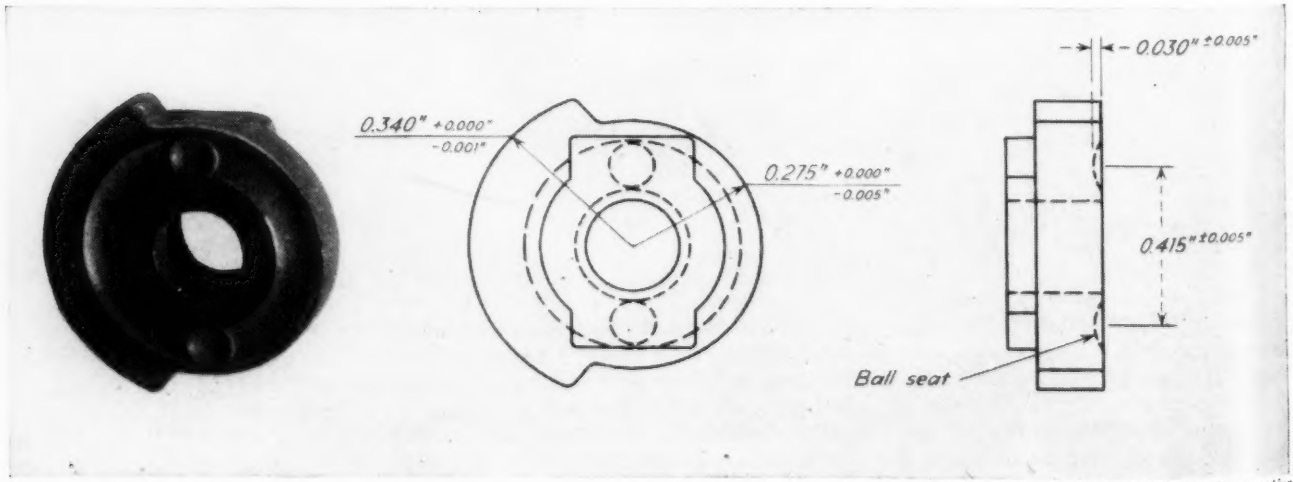
The Presmet Corporation

BRASS LOCK BODY, upper left, is pressed of free machining brass powder. It is machined into the form, lower left, in 12 operations as compared with 26 operations required on the previous cast part, upper right. Other advantages are uniformity of dimensions, consistency of color, and absence of inclusions. Production is 750 parts per hour on one 100 ton press. Cost of the finished part is substantially less than that of a casting.



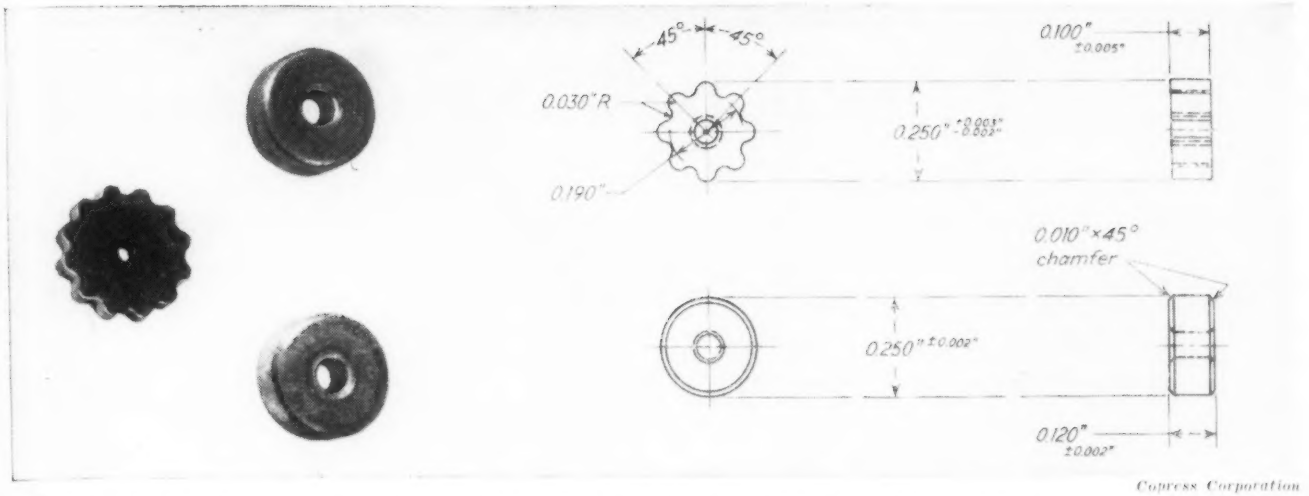
General Sintering Corporation

HUB AND CLUTCH for a phonograph record changer formerly was made in two parts that were assembled by press fitting or staking. The concentricity allowance between the clutch diameter and the $\frac{3}{8}$ in. hole of 0.001 in. total indicator reading made manufacture difficult and expensive. When pressed of powder having an analysis of 90 percent iron, 5 percent copper, and 5 percent brass, a cost savings of 84 percent was realized.



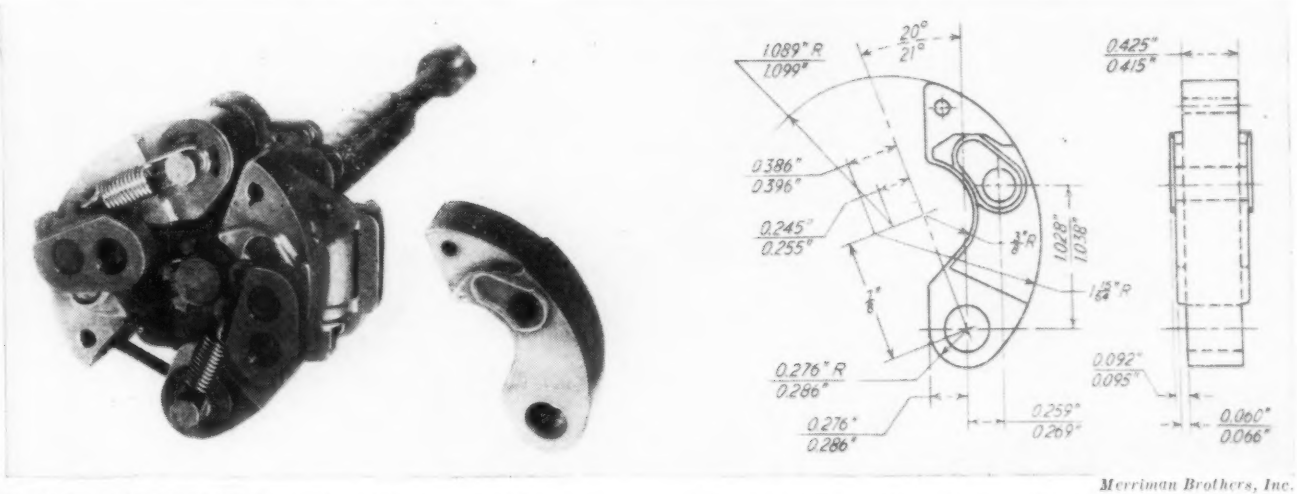
General Sintering Corporation

BEFORE BEING MADE of powdered metal, this ball race consisted of two stampings that were brazed together. Two coining operations were required to shape the stops and ball race. When pressed of 90 percent iron, 5 percent copper, 5 percent brass powder, no subsequent operations are required. Its cost is 80 percent less.



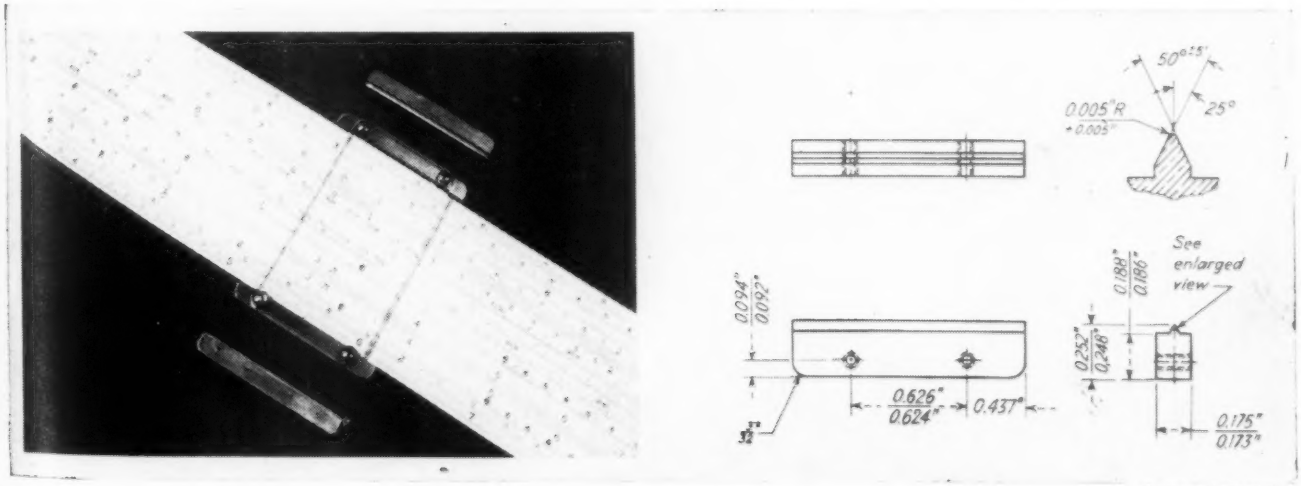
Copress Corporation

THREE REASONS governed the selection of the powdered metallurgy process for the production of these clock parts: Costs were less, chamfers could be obtained without machining, and intricate sections could be produced without difficulty. Both of these parts are sintered of 85-15 brass powder. Shown above are a clock button, center and clock collets. The clock button would be difficult to make by any other method.



Merriman Brothers, Inc.

CLOSE TOLERANCES and high weight were the reasons for specifying sintered metal for this magneto weight. Closer tolerances can be obtained by the powdered metallurgy process than by other methods having a comparable cost; and a heavier part is obtained than would be possible by die casting. The only other practical method of producing these parts to required weight is by precision casting. The material is nickel silver.

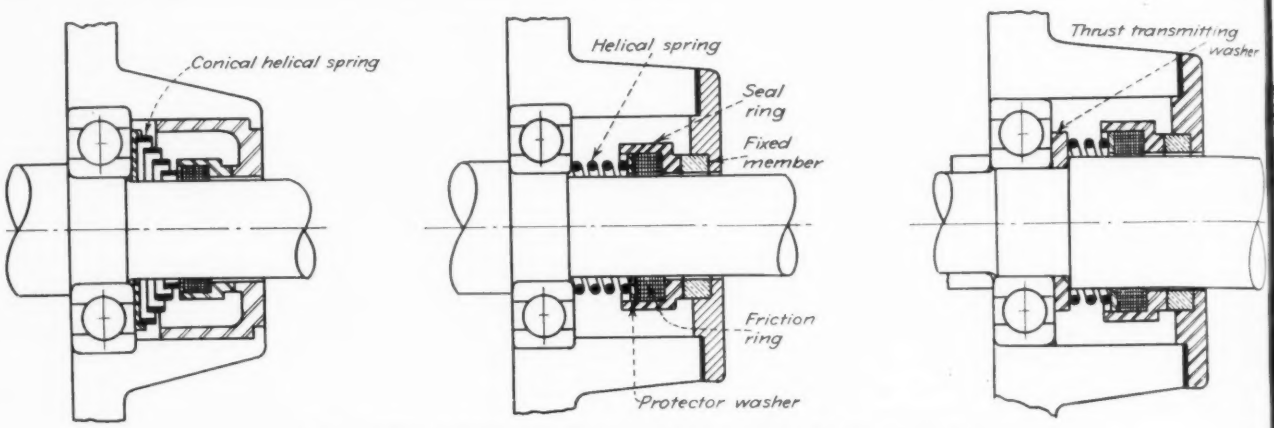


Copress Corporation

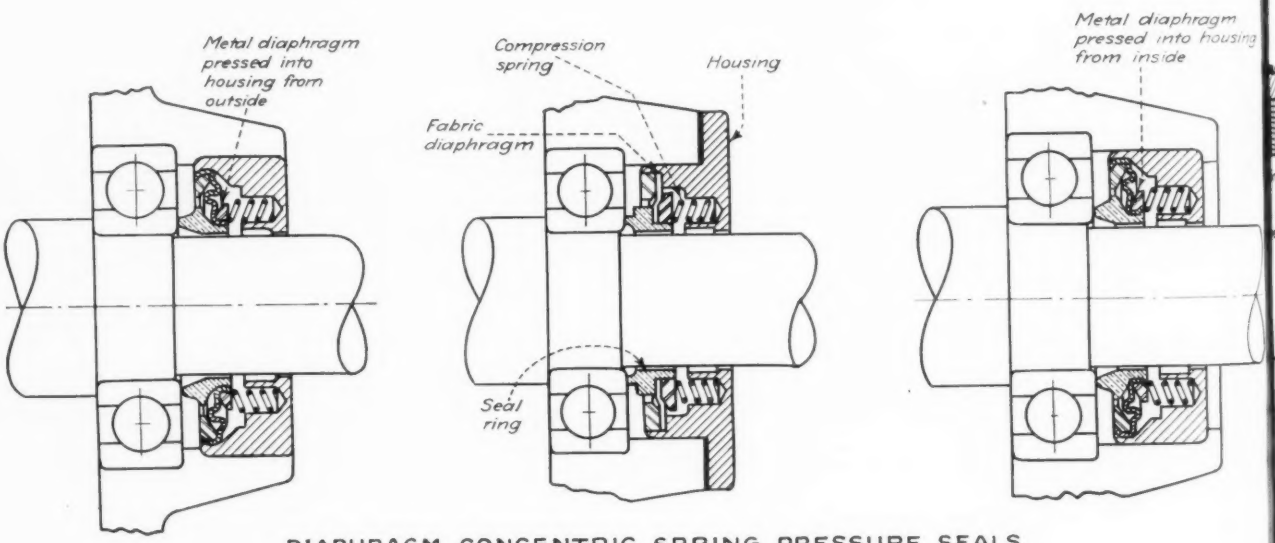
PRESSED of 85-15 brass powder, these slide rule cursor bars illustrated here are made at a rate of 1000 pieces per hour. A cost savings of approximately 63 percent resulted from the switch to powdered metal from machined extrusions. The only disadvantage is that some difficulty is experienced in the plating of them.

Typical Ball Bearing Seals and

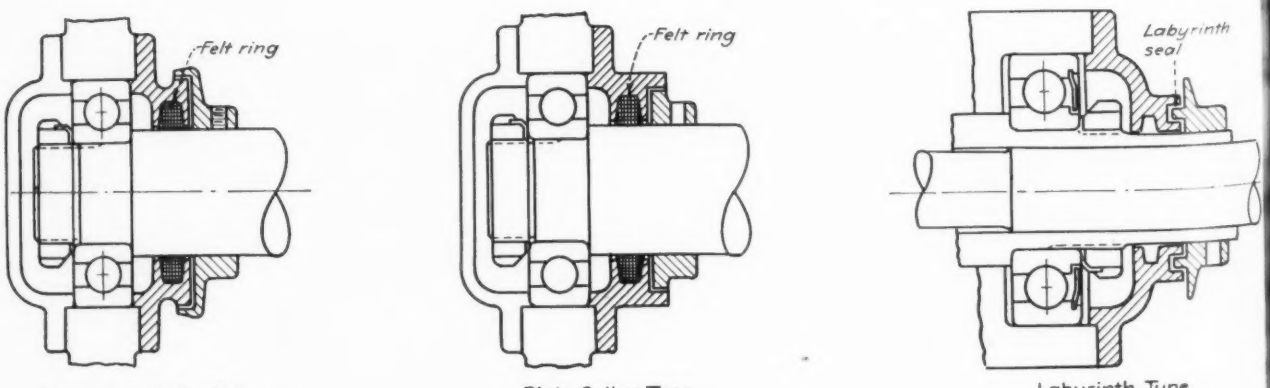
Ball bearings are worthless without provision for lubricant retention and di



ROTARY CONCENTRIC SPRING PRESSURE SEALS



DIAPHRAGM CONCENTRIC SPRING PRESSURE SEALS

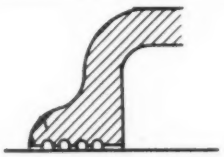


EXTERNAL SLINGERS

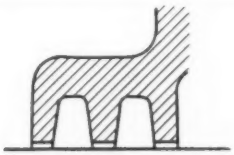
nternal and External Slinger Rings

sistance. Sketches courtesy Marlin-Rockwell Corp. show designs for both functions.

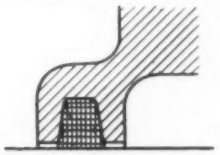
n type of seal, whose tiveness depends on the clearances between bore and shaft



Annular-groove type used to retain grease. Grooves are 1/16 in. to 3/32 in. deep

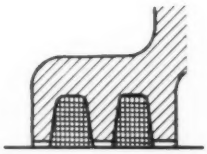


Grease-groove type, to be packed with sealing grease. Suitable for applications where grease is lubricant

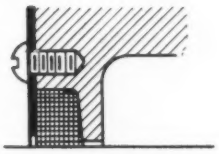


This is the well-known and widely-used felt seal assembled in a machined or cored felt-groove

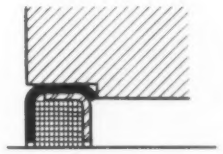
lt and grease groove ration seal. Felt excludes grease-filled groove retains the lubricant



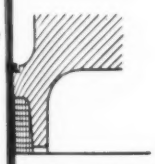
Double groove type of felt seal for positions requiring extra protection against entrance of dirt



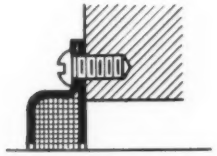
Groove for easily replaceable felt. When felt wears, retaining plate is removed and fresh ring inserted



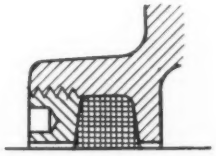
Unit type for pressing in groove in housing. These seals are available in many different shapes



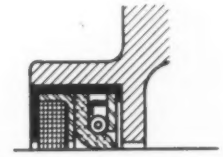
g with pressed steel retaining cap construction permits enient replacement



External type for attaching on outside of housing This is another style of felt seal for easy replacement

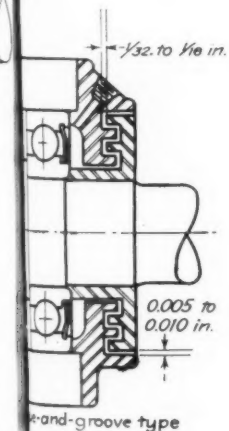


Stuffing box type of felt seal. The felt ring may be somewhat wider in this type. The squeese can be regulated by the nut to suit the compression desired

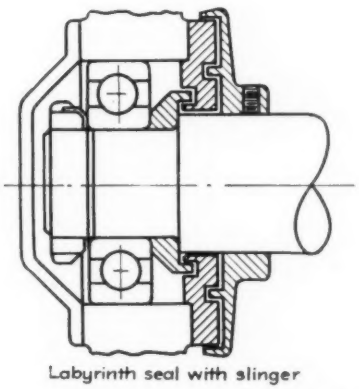


Felt seal used with a leather seal. Felt excludes dirt and leather retains lubricant in this type seal

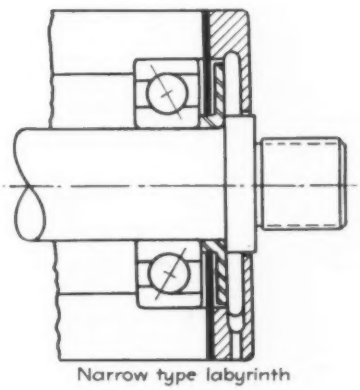
PLAIN, GREASE-GROOVE, AND FELT SHIELDS



t-and-groove type

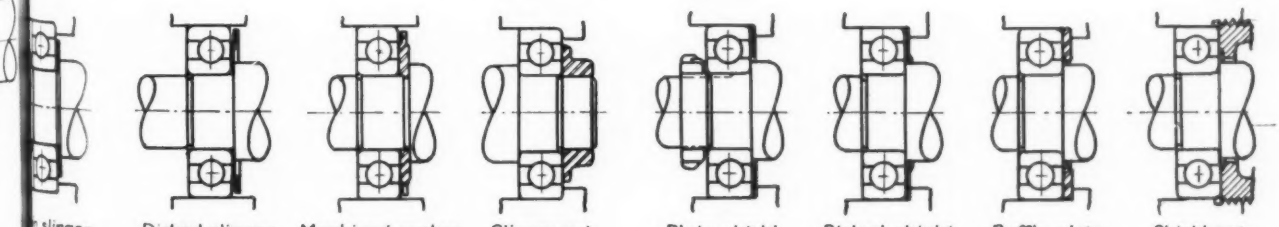


Labyrinth seal with slinger



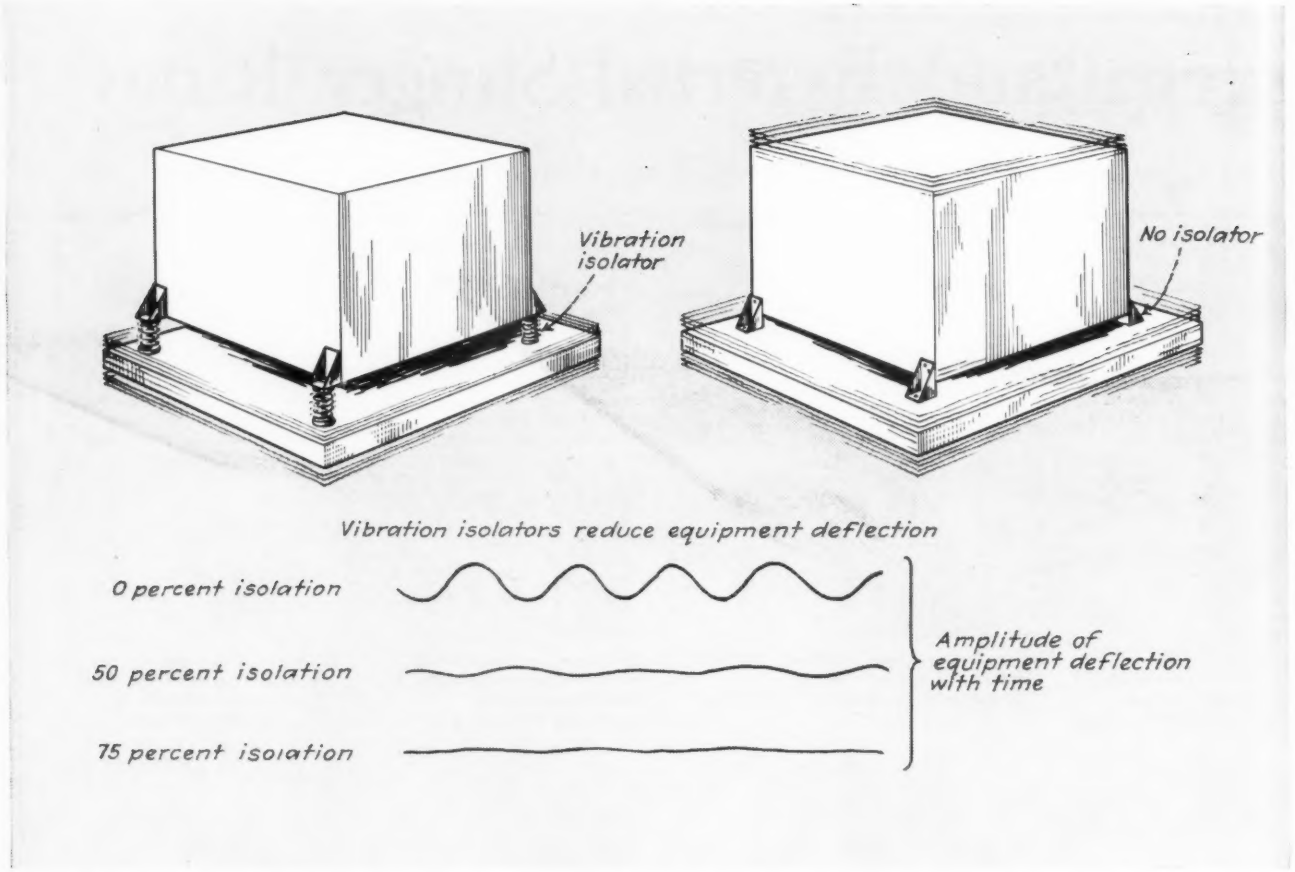
Narrow type labyrinth

LABYRINTH SEALS



slinger | Dished slinger | Machined washer | Slinger nut | Plain shield | Dished shield | Baffle plate | Shield nut

INTERNAL SLINGERS AND SHIELDS



MOST MOUNTS provide less than 100 percent isolation. Above 75 percent isolation, transmissibility ratio is nearly constant.

Vibration and Shock Mounts

Characteristics of materials used and types of available vibration and shock mounts.

CARL J. BASTIEN*
Naval Research Laboratories

THE PROBLEM of determining mounting system characteristics to protect equipment properly from vibration or shock was discussed on page 86 of the September number of PRODUCT ENGINEERING. In this article, application data on mount materials and mount types will be reviewed.

Materials for Mounts

Almost any resilient material can be used, but of these, rubber, cork, felt and metal springs have been used more than others and have proved more satisfactory.

RUBBER is perhaps the best all-around material for general vibration isolation. Many isolators use rubber in shear because the allowable working static deflection of rubber is greater in shear than in compression for equal thicknesses. Rubber in compression should not be used under loads that compress it more than 15 or 20 percent of its original free height; above this, the drift or creep proceeds at an uneconomical rate and the useful life of the mounting is unduly shortened. The characteristics of rubber change with changes in temperature, becoming harder or stiffer as temperature lowers and softer as it rises. It should not be subjected to mechanical work at temperatures above 150 F.

Rubber comes in various hardnesses or durometers. The characteristics of many commercial mounts are varied by changing the metal parts or over-all dimensions. In connection with durometer, it is well to mention that many unpredictable variables combine to determine the effective elasticity or stiffness of a piece of rubber when used as a mounting. If mountings are made from slabs of rubber, it is best to estimate the characteristics desired, purchase the rubber, check the characteristics, and design the mounting based on the individual rubber on hand. When used in shear, the deflection is proportional to load for an extensive range. But this is not true for rubber used in compression.

Rubber, like water, is practically incompressible volumetrically; and if it is distorted in use, space must be provided to allow the rubber to flow. The tension modulus for rubber elongated in the range 0 to 10 percent is approximately 125 to 250 psi. The shear modulus may vary from 55 to 190 psi.

Used in compression installations, the rubber is usually not more than two in. thick. Since rubber ordinarily is not compressed more than twenty percent of its free height, this thickness allows for a maximum static deflection of 0.40 in.

Rubber sandwiches made by bond-

* *The opinions expressed in this article are those of the author and are not to be construed as official opinions of the United States Navy Department.*

ing a rubber slab to two metal plates, and loaded in shear, can be subjected to greater deflections. Those offered commercially by one company have a maximum rated deflection of one inch.

Natural rubber should be protected from oil and may be coated with a flexible varnish for this purpose. Synthetic rubbers are available that incorporate unique features not present in the natural product. Some synthetics possess greater damping or hysteresis than the natural, and are less affected by oil and other hydrocarbons.

An important characteristic of rubber, useful in mount construction, is the ease with which it can be bonded or adhered to most metals. Rubber to metals bonds usually are stronger than the base rubber. Some manufacturers guarantee a bond strength in tension of 200 to 250 psi; however, good design practice limits bond stress to 50 psi. The useful life of rubber mounts, properly designed and installed, is four to eight years. The transmissibility at resonance may be approximately 18 for ordinary compression mountings.

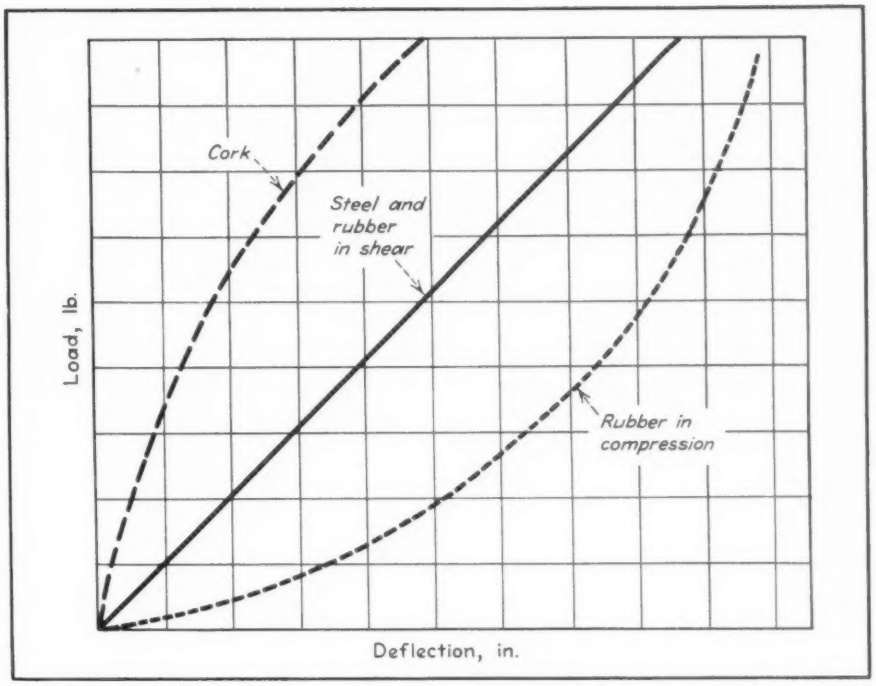


Fig. 1—Load-deflection curves for several common materials used in isolating mounts.

CORK is another material widely used for isolating sound and impact vibrations. Its main use is for supporting large masses, as the entire floor of a building, or large machines resting on concrete blocks, which may be mounted on layers of cork from one to four in. thick. Cork is not usually furnished in its natural state but is finely ground and combined with various binders to obtain more uniform characteristics.

Cork as a vibration isolator is satisfactory only where relatively high frequency vibrations are involved. Oil and water have little effect on it. It may be compressed volumetrically. Long time loading does not destroy the structure or resilience of cork; installations have been known to perform satisfactorily for thirty years. It can be supplied in built-up slabs of any thickness, although two to four inches are the most common. Design static loads for cork in compression may vary from 500 to 30,000 lb per sq ft. In other forms, cork is fabricated into slabs bonded to sheets of ribbed rubber or to some resilient filler material interspaced between strips of cork, the whole covered with asphalt. The transmissibility of cork at resonance is not definitely known, because of variations in its characteristics, but four to eight is a fair approximation.

FELT is made from wool, fur, or hair. It is available in pads of $\frac{1}{4}$ to $1\frac{1}{4}$ in. in thickness. Manufacturers claim that felt is not affected by oil, grease, water, or dirt and will not age or disintegrate under repeated impact pressures. Recommended compressive loads for design vary from 25 to 250 psi for one in. pads, depending on the felt type. According to data supplied by one manufacturer, one inch of felt can provide 80 percent isolation at 20 cps. Felt is reported to incorporate very high damping qualities, making it suitable for use on equipment that must operate through resonant frequencies. Transmissibility at resonance is reported by one manufacturer to be four.

The performance of felt and cork is a function of their manufacture and treating, in addition to an indeterminate quantity that, due to their nonlinear load-deflection characteristics, does not allow definite application of the simple static deflection formula for calculation of the natural frequency.

METAL SPRINGS have many characteristics that make them desirable for use in isolation of mechanical vibrations and are easy to design and fabricate. In the usual installation, springs are loaded in compression, but occasionally they are loaded in shear. Although it is not generally appreciated, horozontal stiffness of springs can be calculated with reasonable accuracy.

Larger working deflections can be safely obtained using steel springs than with any other material, thus, they will isolate disturbing frequencies below the range possible using other materials. They are not affected by temperature changes commonly prevailing.

Springs have two disadvantages: they provide little damping, so that transmissibility at or near resonance may be troublesome, and they allow high frequency vibrations in the sound or noise range to pass. This latter criticism can be overcome if a piece of felt, rubber or other sound absorbent material is used for the spring seat.

Springs are furnished in various forms, but their use is restricted by their high cost, particularly due to the engineering services necessary for individual installations. For use in a product for a delicate or specialized isolation mounting subjected to severe conditions, their use cannot be equalled by other materials. A typical transmissibility at resonance for steel springs is 150. For low frequency isolations, springs may be long and unstable. If deflections over an inch or so are required, the equipment is usually suspended rather than supported from the bottom.

In Fig. 1, typical load-deflection curves for three of the materials discussed are shown.

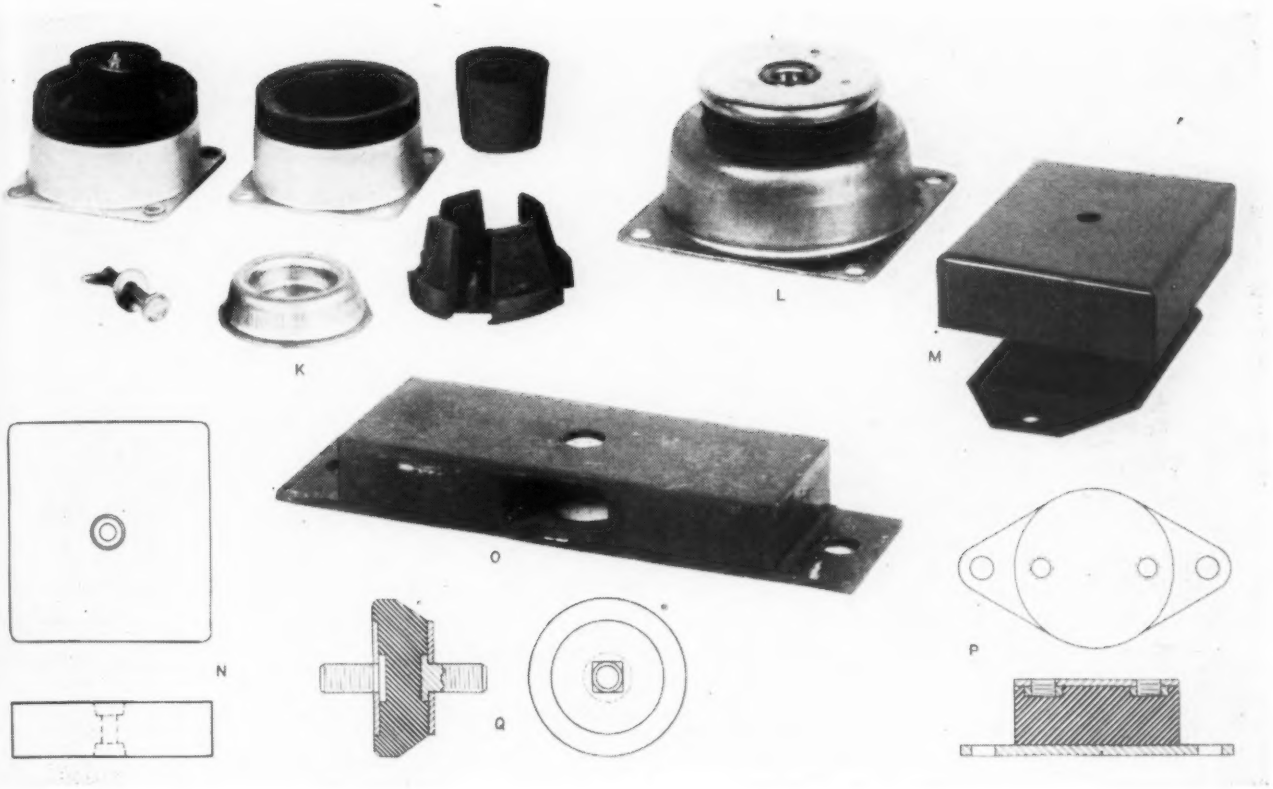
Types of Commercial Mounts

Many types of vibration and shock mounts are commercially available. The most widely used and easiest to apply are made of rubber cased in a steel or aluminum housing. In Table I, a number of vibration and shock mounts are shown with typical applications and other identifying data.

(*continued on next page*)

Vibration and Shock Mounts (continued)

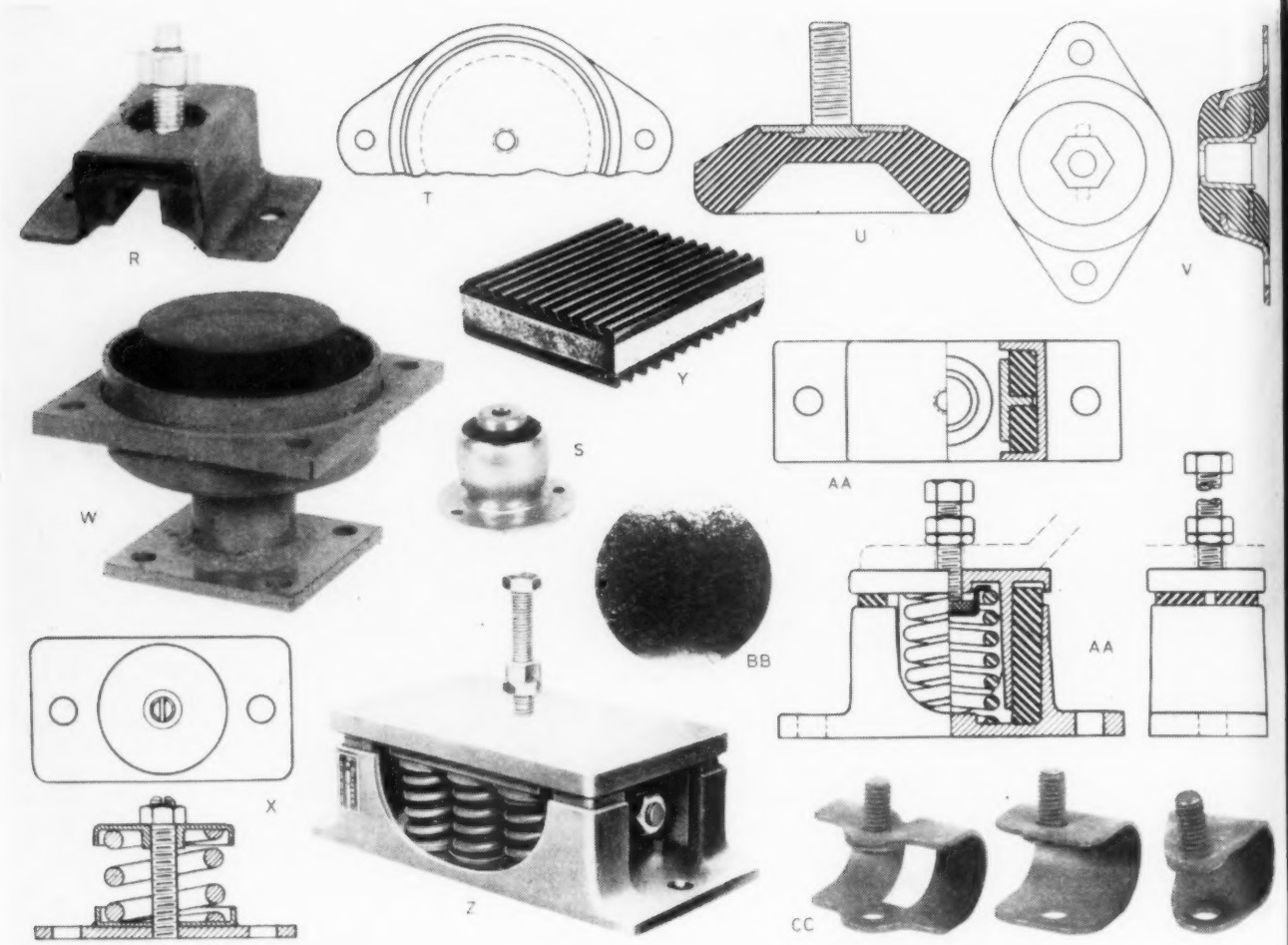
	Resilient Material	Normal Ratings			Over-all Dimensions, in.			Use	Available Commercially	Used for Protection from		Comments
		Load per Mount, lb.	Deflection, in.	Est. Lowest Frequency for 80% Isolation, cps	length	width	height			Shock	Vibration	
(A)	Rubber	50–2670	0.15	20	4 1/2	2 1/2	1 3/4	Commercial equip. Shipboard equip.	X	X	X	Vertical and horizontal stiffnesses equal
(A)	Rubber	50–2670	0.15	20	7 1/2	4 3/4	4	Commercial equip. Shipboard equip.	X	X	X	Vertical and horizontal stiffnesses equal
(B)	Rubber	100–350	0.012	.....	6 7/8	6 7/8	3 9/16	Shipboard	X	X	....	Designed as a shock mount. Equal vertical and horizontal stiffnesses. Resonant over 23 cps
(C)	Rubber	14–150	0.012	.....	3	3	1 1/2	Shipboard	X	X	....	Designed as a shock mount. Equal vertical and horizontal stiffnesses. Resonant over 23 cps
(C)	Rubber	7–60	0.012	.....	2 3/8	2 3/8	1 1/8	Shipboard	X	X	....	Designed as a shock mount. Equal vertical and horizontal stiffnesses. Resonant over 23 cps
(D)	Rubber	1/2–3	0.025±	35±	1/2	1/2	5/8	Commercial equip.	X	....	X	
(E)	Rubber	5–20	0.012	.....	1 3/4	1 3/4	1 1/8	Shipboard	X	X	....	Designed as a shock mount. Resonance over 23 cps
(F)	Rubber	15–55	0.012	.....	2 1/8	2 1/8	1 3/4	Shipboard	X	X	....	Designed as a shock mount. Resonance over 23 cps
(G)	Rubber	1–310	0.06–0.125	30–20	2 1/4	2 1/4	1 3/8	Commercial equip. Shipboard and aircraft	X	X	X	Vertical to horizontal stiffnesses ratio varies. Made with natural or synthetic rubber
(G)	Rubber	1–310	0.06–0.125	30–20	1 1/4	1 1/4	13/32	Commercial equip. Aircraft	X	....	X	Vertical and horizontal stiffnesses are equal
(G)	Rubber	1–112	0.06–0.25	30–15	1 11/16	1 11/16	23/32	Commercial equip. Aircraft	X	....	X	Vertical and horizontal stiffnesses are equal



	Resilient Material	Normal Ratings			Over-all Dimensions, in.			Use	Available Commercially	Used for Protection from		Comments
		Load per Mount, lb.	Deflection, in.	Est. Lowest Frequency for 80% Isolation, cps	length	width	height			Shock	Vibration	
(H)	Rubber	1–112	0 06–0 25	30–75	3 1/4	3 1/4	2 1/2	Commercial equip. Shipboard and aircraft	X	X	X	Vertical to horizontal stiffnesses ratio varies. Made with natural or synthetic rubber
(I)	Rubber	2–45	0 188	17	1 1/4	1 1/4	27/32	Commercial equip. Aircraft	X	....	X	Vertical and horizontal stiffnesses are equal
(I)	Rubber	2–45	0 188	17	2 1/4	2 1/4	1	Commercial equip. Aircraft	X	....	X	Vertical and horizontal stiffnesses are equal
(J)	Rubber	1/2–75	0 125	22	3	3	1 7/16	Commercial equip. Aircraft	X	....	X	Low lateral stiffness
(J)	Rubber	1/2–75	0 125	22	1 11/16	1 11/16	1	Commercial equip. Aircraft	X	....	X	Low lateral stiffness
(K)	Sponge rubber	2–30	0 063	30	2 3/8	2 3/8	1 5/8	Commercial equip. Aircraft	X	....	X	Equal horizontal and vertical stiffnesses
(L)	Rubber	40±	0.063±	30±	3	3	1 1/2	Commercial equip. Aircraft	X	....	X	
(M)	Rubber	1–3800	0 031–0 375	45–12	2 3/4	1 3/8	9/16	Commercial equip.	X	....	X	
(M)	Rubber	1–3800	0 031–0 375	45–12	16	9 1/2	2 11/16	Commercial equip.	X	....	X	
(N)	Rubber	6200–	0 24	15	8	8	2	Commercial equip.	X	....	X	
(O)	Rubber	900–	0.1–0.3	24–14	8	2 1/4	1	Commercial equip.	X	....	X	Can be used in shear or compression
(P)	Rubber	250–750	0.1–0.4	24–12	5	3	1 7/16	Commercial equip.	X	....	X	Can be used in shear or compression
(Q)	Rubber	1–4	0.05–0.2	35–17	3/8	3/8	1 1/4	Commercial equip.	X	....	X	Can be used in shear or compression
(Q)	Rubber	25–75	0.05–0.2	35–17	1 3/8	1 3/8	2 1/8	Commercial equip.	X	....	X	Can be used in shear or compression

(*continued on page 116*)

Vibration and Shock Mounts (continued)



	Resilient Material	Normal Ratings			Over-all Dimensions, in.			Use	Available Commercially	Used for Protection from		Comments
		Load per Mount, lb.	Deflection, in.	Est. Lowest Frequency for 80% Isola, cps	length	width	height			Shock	Vibration	
(R)	Rubber	200	0.25	15	12	4 1/4	1 1/2	Commercial equip.	X	....	X	Furnished in six foot lengths, can be cut as necessary
(S)	Rubber	125–1500	0.3	14	4 1/2	2 1/2	2 1/2	Commercial equip.	X	....	X	Incorporates a high damping feature
(T)	Rubber	125–1500	0.3	14	12 1/4	8 1/4	2 1/2	Commercial equip.	X	....	X	
(U)	Rubber	50–150	0.05–0.3	35–14	2 1/2	2 1/2	7/8	Commercial equip.	X	....	X	Used for table and bench feet
(V)	Rubber	20–2500	0.125±	22–	6 1/4	3 7/8	1 21/32	Commercial equip.	X	X	X	
(V)	Rubber	20–2500	0.125±	22–	3 5/8	2	1 1/8	Shipboard	X	X	X	
(W)	Rubber	100–300	0.015	.....	4 1/2	4 1/2	4	Shipboard	....	X	....	
(X)	Steel spring	50–200	0.2±	17±	4	2 1/4	1 13/16	Commercial equip.	X	....	X	
(Y)	Cork and corrugated rubber	1000–2500	0.1±	24±	12	12	1	Commercial equip.	X	....	X	
(Z)	Steel springs	400–12000	0.1±	24±	18	7	6 3/4	Commercial equip.	X	....	X	
(AA)		75–500	0.1±	24±	6 7/8	2 1/2	3 1/2	Commercial equip.	X	....	X	
(BB)	Rubber coated wool	.....	.....	.....	60	51	1	Commercial equip.	X	X	X	Used for shipping delicate instruments
(CC)	Steel strap	5–15	0.010±	.....	1 1/2	1 1/2	1 1/2	Shipboard	....	X	....	Deforming strap shock mount individually designed

OUTER profile frame for a truck cab. This one-piece part was contour formed on a rotary table from an aluminum alloy.

Rotary Contour Formed Parts

New possibilities for using one piece parts having long, compound curves.

CYRIL J. BATH

President, The Cyril Bath Company

CONTOUR FORMED PARTS enable the designer to use shapes with long sweeping curves having a minimum of joints. These parts often are the most economical way to obtain shapes adapted to modern styling.

Contour forming consists of mounting the forming die on a rotating table; then, depending on the shape to be formed, the material is stretched over the form or rolled on under high pressure. By these methods a formed section produced on rolls or by extrusion can be converted from a straight line shape into various contours. In both kinds of contour forming the outer diameters of the shape are stretched; frequently the inner fibers of the section being formed are compressed simultaneously.

The stretching process is sometimes known as rotary stretch forming. A typical example is illustrated in Fig. 1. This is a high alloy aluminum extrusion contour form to make a bus

Fig. 1—Long sweeping curves of this bus wheelhouse angle are formed by stretching the aluminum alloy section around a die fixed to a rotating table. Three radii in the horizontal plane and five in the vertical are formed without damage to the cross section.

wheel house angle. Three radii are formed in the horizontal plane and five in the vertical plane. By this forming technique the complex shape is quickly and cheaply produced without joints.

Some materials, which will be discussed later, are better adapted to compress forming. This reverse process is performed on the same stretch forming machine by heavy rolls that push the metal onto the die shape on the revolving table. In compress forming, the outer fibers of the member may be stretched; the inner fibers are usually compressed. Compression forming technique applied to car lines for railroad cars is shown in Fig. 2. The parts are of high alloy steel with an elongation of 25 percent and a yield of 50,000 psi. The car lines are sheared, punched and brake formed. The car line is then contoured by holding it against the die with rolls or shoes under constant hydraulic pressure.

Sometimes both compression and stretch methods are required. An example is the motor coach aluminum door frame contour shown in Fig. 3. The severe radii at the top corners are compression formed, and the long radii in the other plane are stretch formed. Forming is done cold. From these illustrations it can be appreciated that contour forming probably is most interesting to the designer because the long sweeping members, made in one piece, offer a simple solution to modern styling demands. If, for instance, automobile manufacturers should design a frame around the outside of the car, the problem could be solved by contour forming the member in one piece, having only one joint. Stainless clad bumpers, formed so that the ends bend around to protect the fenders, can be made in one piece.

Contour forming has these advantages:

1. Special shapes impossible to produce by press forming can be made quickly without heat-treatment or hand work. The process is not competitive with press forming; it merely offers the designer a supplementary range of forms.

2. Joints are minimized because compound curves can be produced in one piece.

3. Little or no scrap results.

4. Clad metals can be used so that the plating operation is eliminated.

5. There is no damage to polished metal faces such as are found on stainless clad material.

6. For some shapes the process is faster than press forming.

7. Aluminum and bronze extrusions, which offer the designer a much greater choice of cross section, no longer are limited to straight line members.

Fig. 2—Carlines, compression formed from high alloy steel having a 50,000 psi yield point. These members are previously sheared, punched and brake formed.

Fig. 3—Severe radii in the top corners of this aluminum door frame are formed by the compression technique and the long radii are made by the stretch forming method.

8. Many materials are strengthened or work-hardened by the application of stretch forming, as long as the material has a fairly high elasticity and the yield point is not exceeded.

What cross-sectional shapes are adaptable to stretch forming? Fig. 4 is a partial answer. Here are shown 26 different cross sections that have been formed successfully on a universal contour former. Undoubtedly there are other shapes that could be contoured successfully.

A second important question is: What materials can be contour formed, and by which method? No inclusive answer can be given this question. The limiting factor is the ductility and the elastic limit of the metal. The degree of ductility and elasticity required will depend upon the sharpness of the curvature desired. Stretch forming is preferred for the high alloy steels. In general, stretch forming is used for the stainless steels because ductility is high and spring back is severe. Other high tensile steels and aluminum are also stretched because the stretch method provides better control of spring back, particularly with constant shallow curves. The low tensile, softer steels are compress formed.

What kind of bends can be made? With the contour method, bends can be either long and shallow, reverse or compound. The limits again depend upon the ductility and elastic limits of the metal. No specific industry-wide formula is known that can be applied to all shapes because of the variation in material and form. In most planes a 360 deg bend is possible with material having proper ductility.

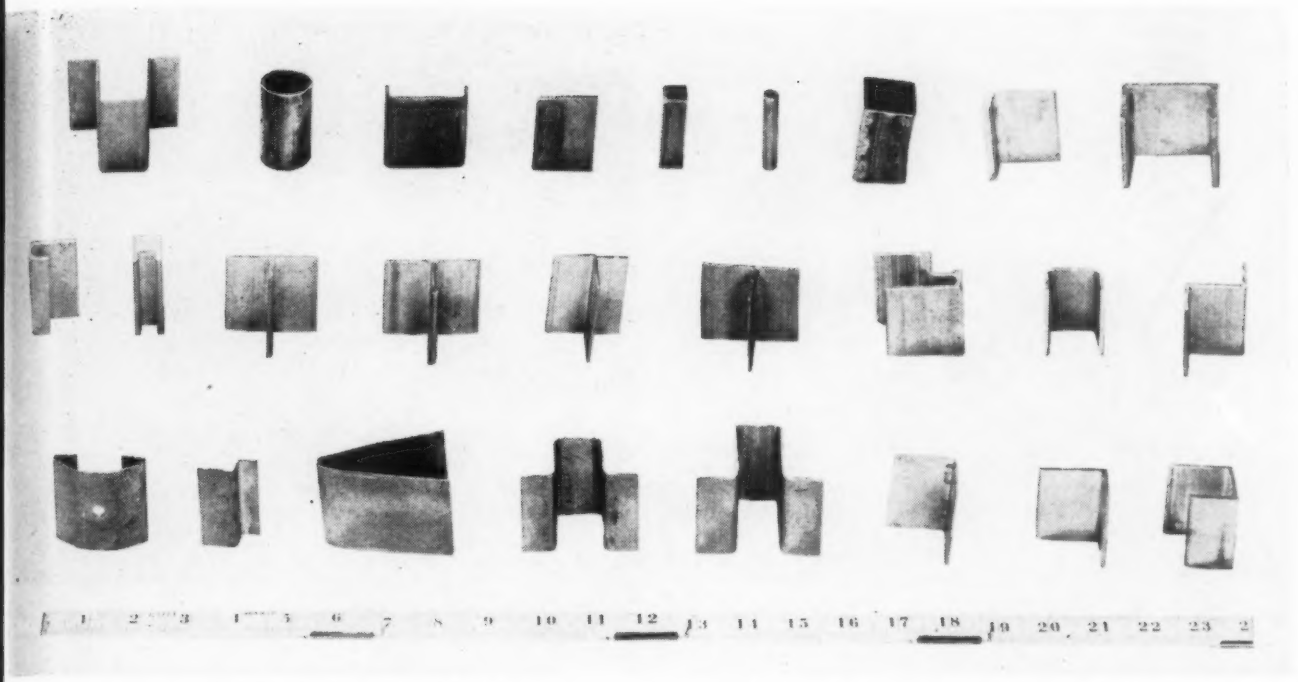


Fig. 4—Structural members in these twenty-six different cross sections have been successfully contour formed on a universal type machine. There is no particular limitation on the cross-section, but the radius of bend is limited by flange depth.

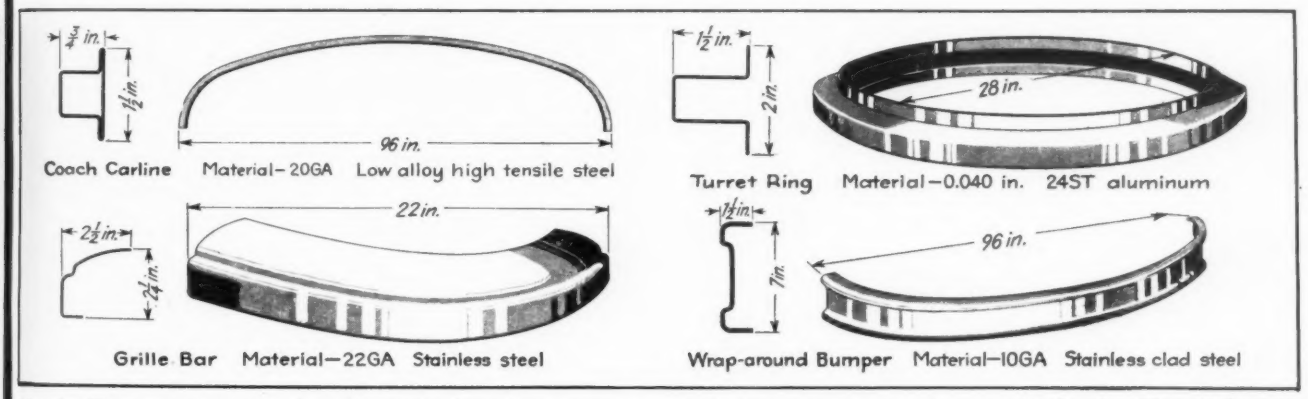


Fig. 5—Flange type structural members that have been contour formed. Comparison of the radius of curvature to the flange depth shows some typical successful bends. The thicker the material, the sharper the bend can be without excessive wrinkling.

Ductility should be such that the difference between yield and ultimate strength is as great as possible. About 20,000 psi is a desirable difference.

No charts or graphs of bends vs thicknesses and springback are available because this new phase of metal working is still under development. Efforts in this direction are not always valid because of the variations still prevalent in metals of the same nominal grade. Therefore, the control of springback is a matter of experience and cut-and-try. By using low-cost dies it is not expensive to rework the die to compensate for the springback after a few experimental pieces are run through the machine. The tolerances that can be held are as good or better than can be had with press forming. For example, aluminum stretch-formed in the ST condition can be held to plus or minus 1/64 in. in a 30 in. sweep of contour.

If sharp curves are required, compress forming is better than stretch forming. On the other hand, the compress forming is not feasible for the stainless steels or the high tensile steels. Sharp bends also introduce the problem of wrinkling. No rule can be given, but the thicker the material the sharper the radius can be without excessive wrinkling. Sometimes pressure is varied in the course of the length of the contour. For instance, if a piece has a sharp radius on the bend and then blends into a broader or softer contour, the pressure will be varied along the length of the member. Problems often occur with flanges. The bare flange generally forms in any shape around a radius that is not less than three times the depth of flange. For example, to form a 1 in. flange requires a radius of 3 in. or more in the usual steels. For softer steels and coppers, a ratio of $2\frac{1}{2}$ to 1 is adequate. With extremely soft metals it might be possible to bring the radius down to $2\frac{1}{2}$ in. on a one-inch flange. For hard steels, on the other hand, the ratio will have to be $3\frac{1}{2}$ to 1. In forming a 1 in. flange with hard steel, a $3\frac{1}{2}$ in. radius is probably the limiting curve.

The thickness of the material does not affect its ratio of flange to radius curvature except that the thicker the material the less wrinkling. A ratio of 3 to 1 of radius curvature to flange thickness is the best guide for the design engineer. Some typical members, Fig. 5, will give the designer an idea of what has been done successfully in different materials.

Another possibility of this technique is sheet contouring. The principal advantage of the rotary method is that

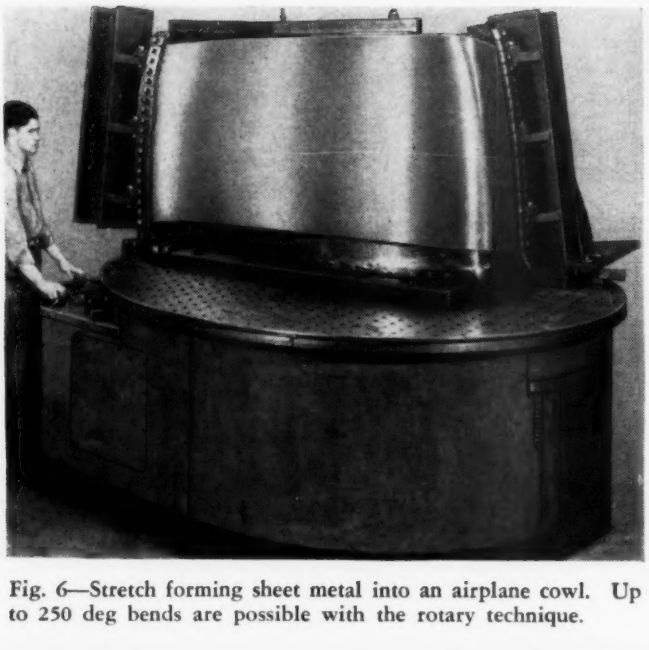


Fig. 6—Stretch forming sheet metal into an airplane cowl. Up to 250 deg bends are possible with the rotary technique.

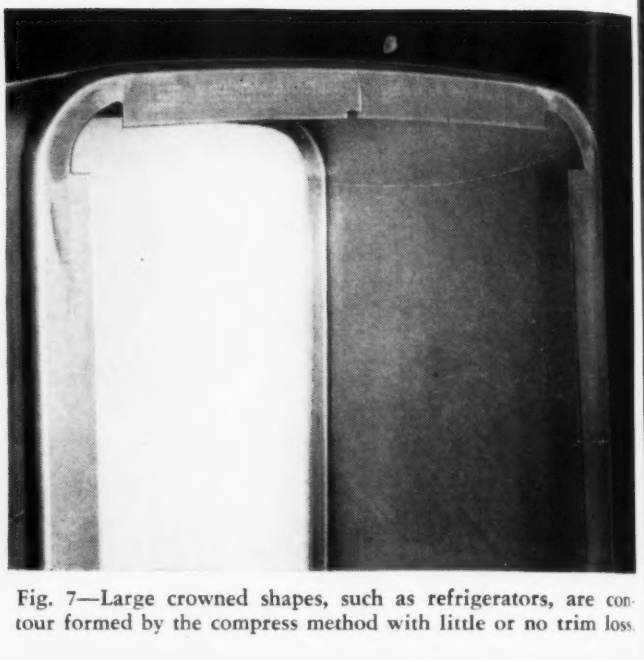


Fig. 7—Large crowned shapes, such as refrigerators, are contour formed by the compress method with little or no trim loss.

Fig. 8—Motor bus bumpers of 10 percent stainless clad, high alloy backing material. Contour forming die costs were cheapest.

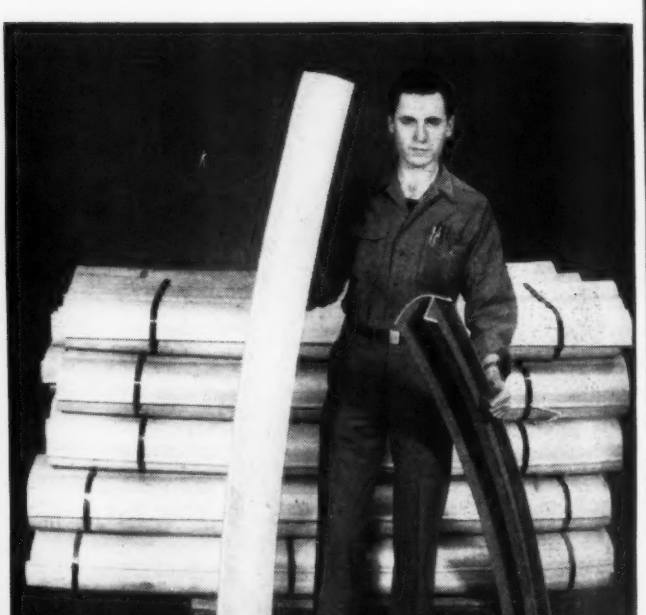


Fig. 9—Top front cowl for a motor bus which was contour formed by the stretch method in spite of a complex cross section.

shapes can be produced up to about 250 deg radius, whereas in conventional stretch-forming methods less than 180 deg is possible. An application of this technique is the manufacture of aircraft cowling. Fig. 6 shows the contour forming of sheet metal.

For radius bends in flange sheets, the limit is dependent upon the amount of stretch which the main body of the sheet must accept to produce the inner radius free of wrinkles. Generally this is done better by the compress method where the flange itself is compressed rather than the body of the sheet stretched. This principle is exemplified in the manufacture of domestic refrigerator cases as shown in Fig. 7. Here it is better not to try to obtain an unwrinkled flange deeper than $\frac{1}{3}$ of the radius. In other words, a cabinet with a 3-in. radius corner can be compress formed with a one-inch flange without wrinkling. If, however, the flange is a triple flange, its depth may have to be reduced to $\frac{7}{8}$ or $\frac{3}{4}$ in., or else the corner radius increased.

It is possible to obtain deeper unwrinkled flanges by special die design. The trouble that arises is that after forming the severe radii is completed, it is likely to leave stress marks running into the remainder of the sheet. In many products this is objectionable. Other examples shown in Figs. 8, 9,

ENGINEERING ABSTRACTS

FOREIGN AND DOMESTIC ARTICLES

Steels for Antifriction Bearings

Foreign abstract, condensed from "Steels for Antifriction Bearings," H. Diergarten, *VDI Zeitschrift*, March 21, 1942, page 167.

STEELS FOR ANTIFRICTION BEARINGS belong in the group of chrome alloy steels. They are distinguished by their capacity to resist highly localized tension-compression and wear stresses. Variations in microstructure are not permissible in such steels.

The requirement of uniformity refers not only to the analytical composition, microstructure, grain size and freedom from slag inclusions, but it also includes good machinability of races, "pressability" of balls and rolls; and the regularity of quenching, tempering, grinding and polishing qualities.

Quality of steels for antifriction bearings depends largely on the raw materials, the type of furnace and subsequent treatment. Such steels are produced in acidic or basic open hearth processes or in electric furnaces. Sand or slag inclusions must not occur. The quality must be such that carbides are not torn out by grinding operations.

The density of these steels lies between 7.7 and 7.8 grams per cu cm depending upon the chrome content.

Directly hardenable chrome steels, having a low overeutectoid carbon content to increase wear resistance, constitute the majority of steels employed. Uniformly high hardness and toughness are secured by addition of 1.5 percent chrome, which also promotes fine grain size. Oil hardenability of sections as large as 30 sq mm is improved by this addition, which also improves tensile and compressive strength qualities. For larger sectional areas it is advantageous also to increase the manganese and silicon content.

All the steels in the first four rows of Table I have the same manganese and silicon content, but vary with respect to the chromium and carbon content in such a way that the chromium content increases with increasing dimensions of the parts, while carbon content decreases, to maintain a fine distribution of the carbides.

It has been found that addition of molybdenum and vanadium does not improve the quality of the steel, but is detrimental because the tendency to crack is increased with increasing molybdenum content.

Where the specific pressure is less severe, as in large races of low hardness and in spring roller bearings for agricultural machinery, the composition of the steel is modified. (This refers to Mn-Si or Mn-Cr spring steels with about 0.35 to 0.50 C.)

Corrosion resistant antifriction bearing parts can be produced from two different groups of materials. First, steel with 0.4 C and 15 Cr for hardened parts with a Rockwell hardness of 55R_c is sufficiently corrosion resistant when hardened and polished. When necessary it is possible to exceed a hardness of 55R_c by the addition of small quantities of molybdenum, cobalt or manganese while simultaneously increasing the carbon content. But such additions will reduce corrosion resistance and machinability.

Non-hardenable steels, on the other hand, with 18 Cr and 8 Ni possess a greater corrosion resistance even when not polished, but their hardness is relatively low (about 200 BNH). Parts made of this material must necessarily be of larger dimensions for a given bearing life to compensate for the low hardness.

The steels listed in Table II are occasionally used, however, they do exhibit sufficient load capacity and endurance to replace the steels listed in Table I, they are also more difficult to produce.

Steels for antifriction bearings are marketed as annealed bars, wire, tubes, rings and disks. In Table III are

Table I—Chemical Composition of Directly Hardenable Steels

Application	Elements, Percent						
	C	Cr	Mn	Si	P, max	S, max	P+S, max
Balls, rolls and needles not larger than 10 mm dia	1.0 –1.2	0.40–0.60	0.25–0.40	0.25–0.35	0.03	0.03	...
Balls and rolls not larger than 20 mm dia	0.95–1.15	0.80–1.20	0.25–0.40	0.25–0.35	0.03	0.03	...
Balls and rolls larger than 20 mm dia, standard races and disks	0.90–1.10	1.30–1.70	0.25–0.40	0.25–0.35	0.03	0.03	...
Races and disks having a section larger than 30 by 50 mm	0.90–1.05	1.50–2.00	0.25–0.40	0.25–0.35	0.03	0.03	....
Large ball bearing races of tempered steel	0.30–0.55	1.0 –2.0	0.25–0.50	0.25–0.40	0.04	0.04	0.07
Large ball bearing races of tempered steel	0.45–0.65	.....	0.90–1.40	0.70–1.30	0.04	0.04	0.07

Table II—Chemical Composition of Case-Hardening Steels

	Elements, Percent								
	C	Mn	P, max	S, max	Si	Ni	Cr	Mo	V, min
Mn-Steel	0.25–0.35	0.50–0.90	0.04	0.04	0.25–0.35	.....	.....	.....	...
Cr-Steel, SAE 5120	0.15–0.25	0.30–0.60	0.04	0.05	0.15–0.30	.....	0.60–0.90	.....	
Ni-Mo-Steel, SAE 4615	0.10–0.20	0.40–0.70	0.04	0.05	0.15–0.30	1.65–2.00	.....	0.20–0.30	....
Cr-Ni-Steel, SAE 3115	0.10–0.20	0.30–0.60	0.04	0.05	0.15–0.30	1.00–1.50	0.45–0.75	.....	....
Cr-V-Steel, SAE 6120	0.15–0.25	0.30–0.60	0.04	0.05	0.15–0.30	.....	0.80–1.10	.....	0.15

given standards adopted for base materials.

Lamellar cementite in the microstructure should be avoided, because it reduces machinability and causes trouble on press operations. On the other hand, when the raw material is too soft machinability is poor and it is impossible to produce a satisfactory surface and finish.

Antifriction bearing steels are usually annealed at a temperature somewhat higher than the point of transformation, that is, at 730 to 750 C depending on the chromium content. When annealing at 750 to 780 C and cooling slowly in an oven at 10 to 15 deg C per hr, it is possible to produce granular cementite and a soft microstructure at a faster rate than when annealing at 730 to 750 C. Hardened parts are annealed for two hours at 700 C. If a second hardening is necessary it is advisable to normalize by maintaining a temperature of 650 C for one hour.

Standard balls and rolls are usually quenched in salt or sweet water; small balls, rolls and needles in mineral oil of 2½ deg Engler viscosity based on the 20 C standard. The temperature varies between 780 and 840 C and depends upon the dimensions of the parts and on the alloy content.

Races and disks are quenched in agitated mineral oil of viscosity 3 to 8 deg Engler (at 20 C standard) from a temperature of 830 to 880 C. The parts should always be well heated to loosen the carbides.

Parts must be tempered at 140 to 160 C (after quenching) for one to two hours under average conditions. Higher temperatures and longer times are used in special cases resulting in greater toughness and reduced hardness, which is equivalent to a reduction in the life of the bearing.

Since hardness and toughness in conjunction with composition of the steel determine the load capacity and life of antifriction bearings, it is advisable to use the standards given in Table IV as a basis for ascertaining normal bearing life.

A rapid decrease in bearing life occurs when Rockwell hardness is less than 58 (about 600 Brinell); while hardness, toughness and steel quality supplement each other in the range between 58 and 60 Rockwell (600 to 700 Brinell) in such a way as to maintain the bearing life independently of the hardness variation within these limits.

The surface condition of the parts, based on long time experience, is also indicated in Table IV. A higher finish than suggested may often result in dimensional inaccuracy caused by a lengthy polishing operation, which should be avoided.

Table III—Condition of Annealed Steels as Delivered

	Dimensional Accuracy	Tensile Strength, kg per sq mm	Elongation for $l=10d$ percent	Brinell, kg per sq mm	Comments
For balls, rolls and needles, drawn in rings or bars	DIN 668, or according to agreement.	62–85	12–22	180–250	Decarbonization and scratches must be avoided.
Black bars and tubes	According to agreement; stock of 3 to 6 mm on finished dimension.	60–75	15–22	170–210	Finished surface without defects.
Premachined bars and tubes	DIN 668, or according to agreement.	60–75	15–22	170–210	Finished surface without defects.
Rolled or forged rings and disks	According to agreement; stock of 4 to 15 mm on finished dimension.	60–75	.....	170–210	Particular care required in heat-treating heavy race sections with thin flanges.

Table IV—Hardness and Surface Condition of Finished Parts

	Rockwell Hardness	Surface Condition
Standard races and disks.....	58–66	Polished grooves in race; mother surfaces finish ground.
Races of tempered steel.....	Tensile strength 100–120 kg per sq mm	Polished grooves in race; mother surfaces finish ground.
Balls.....	62–68	Polished.
Rolls.....	58–65	Cylindrical surface lapped; end surfaces finish ground.
Long rolls or needles.....	55–62	Cylindrical surface lapped.
Spring rolls.....	40–50	Fine finish ground.
Long rolls for spring roller bearings	48–58	Fine finish ground or lapped.
Split shells.....	42–52	Drum shined.

Acrylic Plastic Light Fixtures

From "Lighting Fixtures from Acrylic Sheet" by F. W. Tetzlaff, Rohm and Haas Company, presented at the annual meeting of the Society of the Plastics Industry, May 21, 1948, Atlantic City, N. J.

ACRYLIC SHEET MATERIALS are finding a widening market in the lighting fixture field. Interest has been aroused because of the special properties inherent in the acrylics. The acrylics are generally available in the form of sheets, rods, and tubes, as well as molding powder for injection or compression molded parts. The two principal producers of these materials are the Rohm and Haas Company and the E. I. duPont de Nemours Company. In addition to the clear material, which has a light transmission of 92 percent, a wide variety of transparent, translucent, and opaque colors is also available. When desirable, the sheet can be made in a number of different patterns as well as in corrugated form.

For fluorescent shielding, the material most commonly used has been one of the available white translucent grades. This material has excellent light diffusing properties and, in addition, absorbs practically no light since the total of the transmitted light plus the reflected light equals approximately 100 percent. This characteristic is important because it makes possible better lighting with lower electrical input than with other plastics, which absorb some light.

When the lighting engineer requires a material which has a high reflectance value, but also has some transmission, there is available a denser material than the translucent, particularly adapted for use where low fixture brightness is desired.

Advantages of importance are excellent outdoor weather resistance as well as lack of discoloration upon pro-

longed exposure to fluorescent light. Laboratory tests (ASTM D795-44T) show the white translucent material to be superior in this respect. Such discoloration causes efficiency of the fixture to be radically reduced because the transmission values are appreciably lowered. Outdoor weather resistance of the acrylics has been well proven.

Because of its light weight and easy formability, a relatively large unsupported area can be covered with a single piece of material. Recently fixtures eight feet long have been formed for use as diffusing shields with fluorescent lighting. The availability of long sheets make possible a reduction in the actual number of fixtures required when this is advantageous. Large sections have been used for walls and murals, to take advantage of its light "piping" qualities.

The excellent dimensional stability of the acrylics when sections are properly fabricated overcomes former objections to the use of plastics for lighting. The Rohm and Haas Company has on test, for over eight months, two reflectors approximately four feet long and 12 in. wide. These have been operated continuously at an ambient temperature of 77 F. Approximately one month after they were placed in service, a change in contour of $\frac{1}{32}$ in. was noted, and, since that time, no other measurable change has taken place.

The shatter resistance of these materials is well known. Over a wide temperature range a marked superiority over glass is evident. In addition, the acrylics do not become brittle at reduced temperatures. Following are data on falling ball impact tests:

When compared to plate glass or double strength window glass, the acrylics in 0.125 in. thickness have an impact strength of 3.5 ft-lb compared to 0.5 ft-lb for glass; in 0.250 in. thickness, the acrylics have an impact strength of 9.5 ft-lb compared to 0.7 ft-lb for glass. The impact strength of the acrylics increase with increasing thickness, regardless of the type of impact.

The most common objections to the use of the acrylics in the lighting field have been:

1. Higher price than other materials.
2. Static attraction or dust accumulation.
3. Lack of abrasion resistance.

The most important deterrent to large volume-increase sales in this field has been the price factor. However, as less expensive materials are found wanting in specific requirements, the acrylics will find even more general use than at present.

The static attraction difficulty is one common to those materials having good electrostatic properties. The development of excellent anti-static waxes recently has done much to alleviate this problem. Waxes are now available which are easy to apply and which have been of material assistance in making maintenance simple on all types of acrylic parts. The scratching objection is of minor importance as long as harsh cleaning agents are not used on acrylic lighting fixtures, since the fabricated sections usually are installed in locations where they cannot be easily abraded.

Probably the most widespread use in this field has been in railway car lighting for a number of years. High impact strength and a resultant ability to withstand repeated shocks without failure have been important factors in the specification of acrylic sheeting. Another use where acrylic sheet has been employed extensively is in sign tracks. This application is both a sign and lighting use.

To reduce high maintenance costs, municipalities and maintenance companies have used substantial amounts of acrylic material for street lighting purposes. Both clear and patterned material have been used for this application. The use of the patterned material gives good hiding of hot spots while maintaining maximum light transmission. The patterned surface is used on the outside of the globe to obtain the best transmission possible, and to avoid flattening during forming.

Experiments with electric street lamps indicate that the acrylics can be used with lamps up to 6,000 lumens if the ventilation and design is proper.

Modern Dimensioning Practice

From the paper "Modern Dimensioning Practice" given by S. B. Elrod, Assoc. Prof. Purdue University, at the fifty-sixth annual meeting of the American Society for Engineering Education, Texas University, June 14, 1948.

ALTHOUGH DIMENSIONING PRACTICE is the result of one hundred and fifty years of evolution, the greatest strides in development have taken place in the last two decades. Before the first world war only a small minority of industries had progressed beyond the stage of custom manufacturing. Only basic dimensions were put on a drawing, and little effort was made in selecting these. In many cases the only tolerances used were ± 0.001 in. for close work and ± 0.005 in. for rougher work, used along with the old expressions, "Slip fit," and "Drive fit."

In contrast some industries have gone to great lengths to work out a system of dimensioning that leaves little to the judgment of the shop man; the sole responsibility for the functioning of parts rests with the engineering department. Every dimension on the drawing carries a specific tolerance or is expressed as a limit dimension. And finish is specified for every surface.

Dimensioning practice has as its objective, the ability to produce a complete set of separate working drawings for a large number of complex parts, each of which is to be manufactured by a separate organization, with no liaison; the ability to use all parts so produced, with no selection, in the assembly of mechanisms, all of which will function properly and efficiently and at the same time keep drafting time and effort to a minimum.

Decimal Dimensions

Fractional dimensions are giving way to decimal dimensions. The two place decimal system inaugurated by the late Carl E. Johanssen, in the Ford Motor Company plants, has gained ground rapidly and is used by most of the automotive industry and over 80 percent of the aircraft industry. The system has all the advantages of the metric system without requiring a re-educational and re-tooling program. Surveys conducted periodically show the shift to decimal dimensions continues steadily.

Functional Dimensions

In the past it was assumed that the processes involved in the manufacture of a part were the governing factor in dimensioning it. However, the overriding need for part performance and interchangeability has invalidated that assumption; for most cases the proper functioning of the part is paramount.

In the new proposed American Standard, "Principles governing design and dimensioning with application of tolerance and allowances for interchangeable manufacture", the following statement is made. "In examining the project, to determine its

suitability for economical manufacturing, inspection and assembly requirements, primary datum surfaces must be established from functional considerations."

Ordnance Manual, Arts. 380.01, 380.02 and 380.03 read as follows: "Dimension and tolerance surfaces in the order of their importance in the functioning of the mechanism. The relationships left without direct dimensioning and tolerancing should be those of least importance. Dimension directly to the surfaces which make contact with other parts in the operation of the mechanism. Having placed dimensions and tolerances in accordance with functional requirements, study the design to see whether any rearrangements would make machining faster or less expensive. On ordnance drawings do not rearrange dimensions and tolerances for machinability unless this can be done without materially affecting the accuracy and utility of the design."

Current Systems

Current systems for dimensioning for interchangeability differ only in detail. Three of these systems have appeared as follows:

1. "SAE Aeronautical Drafting Room Manual," published in 1946 by the Society of Automotive Engineers. This is a cautious transition compared to the effort of several other agencies.

2. "Drafting Room Practice in Relation to Interchangeable Components," by C. A. Gladman, Scientific Officer, National Physical Laboratory, England. Published by the SAE in 1945, this treatise, known in drafting circles as the "Gladman Paper," stirred up quite a bit of controversy on two counts; it is not easy to read or digest and would require complete shop re-education if the practices contained in it were applied.

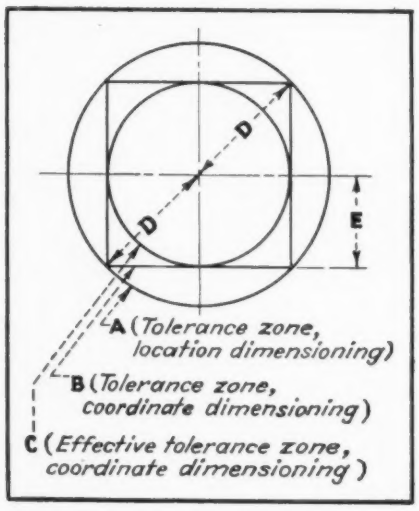


Fig. 1—Comparison of tolerance zones produced by location dimensioning and coordinate dimensioning.

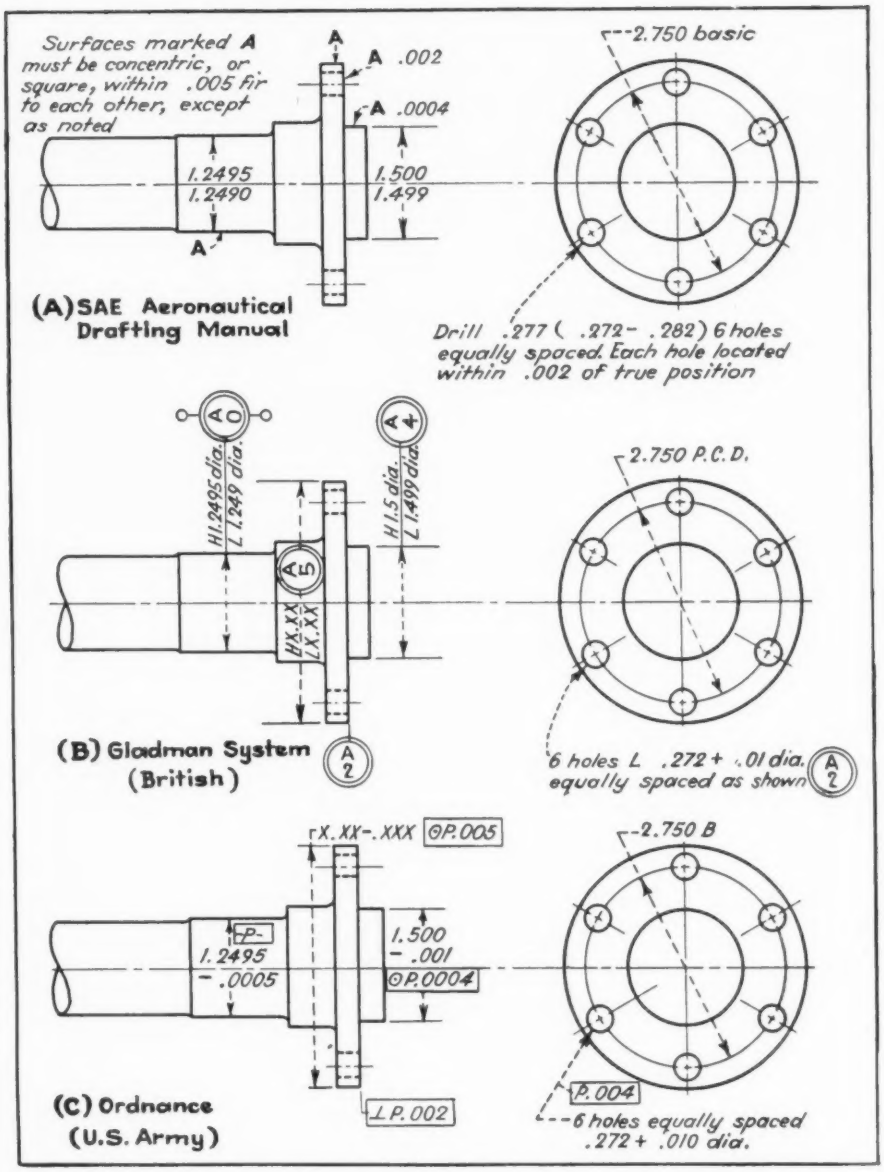


Fig. 2—Flanged part drawn up to conform with three location dimensioning systems.

3. "Ordnance Manual on Dimensioning and Tolerancing," by the Inspection Gage Sub-Office, Office of the Chief of Ordnance, A.S.F., United States Army, issued in 1945. This is much more comprehensive than the other two manuals listed.

All three systems appearing in these manuals, are different approaches to the locational-tolerancing system. The underlying philosophy is: Remove all tolerances from location dimensions and allow variations to exist in the location of elements relative only to their theoretically exact location.

If all location dimensions are considered to be basic, without any tolerances, the effect can be illustrated by the enlarged diagram in Fig 1 (Ordnance Manual). The two intersecting centerlines represent the theoretical location of a hole or pin, or other circular element.

If design allows the center of a particular hole or other element to vary from the calculated center by an amount D, then the center may lie anywhere inside the area of the larger circle A of $2D$ diameter. If this same element is located by using the conventional coordinate system, the designer reduces the allowable variation to $E = \frac{D}{\sqrt{2}}$, which is the tolerance for coordinate dimensions. This reduces the area of possible variation from πD^2 to $2D^2$ or a reduction of 36.6

percent as shown by the inscribed square *B*. When gages are made to check the location of this element it is necessary to further restrict the location of its center to the area of the smaller circle *C* inscribed within the square; a full 50 percent reduction of the tolerance area. The difference in the two methods has production importance and definitely favors the locational-tolerancing method. Further, this system is easier to use in the drafting room. For parts conventionally dimensioned, problems of tolerances, allowances, etc., are complicated by introduction of trigonometric values between holes and design elements.

Shown in Fig 2(A) is a sample part drawn in conformance with the SAE manual. The practice of calling for holes equally spaced on a bolt circle is used here for simplicity only; a set of coordinate dimensions would be required in many places. The diameter of the bolt circle marked "2.750 basic" means exactly 2.750 in. with no variation allowed. The term, "equally spaced," then locates the theoretical centers of these holes. This implies no error which is impossible to realize in practice. However, the note, "each hole located within 0.002 of true position," allows considerable leeway in that it is only necessary to locate holes within that distance of theoretical center, or the center of a tolerance circle of 0.004 dia. If the center of the holes lie within the tolerance circle the part is acceptable and will function correctly.

The Roman letter *A*, with or without accompanying values, taken together with the typical note, control the trueness of some of the surfaces. Surfaces marked *A* (no accompanying value) may be eccentric 0.005, Full Indicator Reading, as indicated by the general note. The flange face, marked *A*.002, must be square and true within 0.002 FIR, and the pilot diameter is allowed an eccentricity of only 0.004 FIR. No indication is made of the preferred dimension in this system. All dimensions are given in the form of limits and production can choose within these limits.

Gladman System

The same part is shown in Fig 2(B) drawn to the Gladman system. The note calling out six holes in the flange is briefer than the SAE note. Symbol *A*/2 inside the double circle reads as, "each hole within 0.002 of true position with respect to datum surface *A*. Datum surface *A* is identified by *A*/0 in the circle with the dumbell. This is the same as the SAE system except for the reference to a datum surface and the omission of the word "Basic."

The method of indicating dimension and tolerance values is involved. High and low limit dimensions are shown preceded by the letters *H* and *L* and the interpretation is exactly the same as our limit dimensions; the shop may choose between these limits.

Ordnance Manual

The same part is shown in Fig 2(C) dimensioned accorded to the Ordnance Manual. The result is the same but there are several features that deserve considerable attention. Whereas the Gladman system used the same symbol to control all relations, the Ordnance system modifies the symbol for each case. The *P*.004 inside the rectangle controls the location of hole relative to each other and to the datum surface *P*. Concentricity, squareness, parallelism are indicated by added symbols, two of which are shown on the drawing. Parallelism is indicated by two parallel lines, the datum surface letter and the tolerance, all in a small rectangle.

All dimensions on Ordnance drawings are in basic form. If tolerances apply, only one unilateral tolerance is given. If no tolerances apply, as for the basic location dimension, the dimension is followed by the letter *B*. The basic size is preferred but may vary by the amount shown for toleranced dimensions.

An outstanding item is the interpretation of the note locating the holes in the flange. The symbol *P*.004 means the same as the Gladman symbol *A*/2. But the Ordnance Manual goes further than this by considering the method of gaging such parts and their utility. In this case, the six holes would be checked with a gage having six pins of 0.272-0.004 or 0.268 dia, equally spaced on a 2.750 dia circle located relative to the shaft bearing diameter. This means that holes of just 0.272 dia can vary only by the amount specified but that oversize holes may be still further away from the true position and still be accepted by the gage. If the holes are a full 0.010 oversize, the diameter of the tolerance circle can be increased by the same amount, $0.004 + 0.010 = 0.014$, and the part will still function properly. Any variations in individual holes, out of round, axis not parallel, would reduce allowable variation.

Developments in Recording Oscillograph Traces

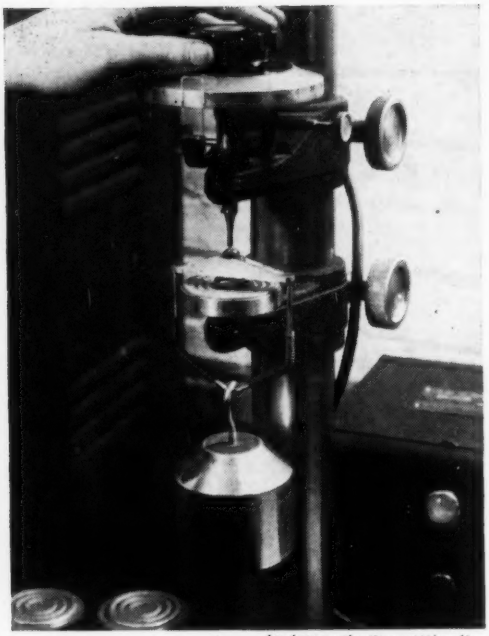
Foreign Abstract from "New Developments in Recording Oscillograph Traces" in *Electronic Engineering*, London, February 1948, Vol. 20, No. 240, p 58.

IT IS WELL KNOWN that high speed mechanical or electrical variations can be translated by a standard commercial oscillograph into visible traces on a fluorescent screen. The impermanence of the traces, however, makes difficult any accurate study of the inter-relationship of several phenomena and makes impossible any subsequent checking. Attempts have been made with varying success to photograph these traces by means of standard cameras.

The number of traces that can be so recorded simultaneously is limited in practice by the space between the screens. This necessitates moving the camera back so far that the traces are reduced to an impracticable scale. To provide a record on a practical scale where a number of traces are required to be shown side by side, Messrs. Avimo, Ltd., now have in production a series of recording cameras with built-in cathode ray tubes, the latter so arranged that their traces are photographed through a mirror. In this way, as many as 15 traces plus a timing interval can be recorded on a scale adequate for practical purposes.

In one form of apparatus, designed for the Ministry of Supply, the outputs from a series of pick-up units are amplified and recorded on six cathode ray tubes $1\frac{1}{2}$ in. in dia built into the camera.

The maximum length of trace on each tube screen is 1 in., which is reduced to 10 mm on the film. Provision is made for the use of standard 70 mm perforated film or paper in 100 ft lengths wound on metal spools. A microscope is provided which permits viewing of the traces while recording is in progress. The cathode ray tubes are each mounted in a mu-metal shield, so arranged that the light from the tube heaters has a minimum effect on the film. It is possible by an adjustment, which is not affected by normal vibration and handling, to orient each tube in its mounting. The traces lie on one straight line on the record, at right angles to the direction of film travel. A range of six film speeds is obtainable: approximately 1, $2\frac{1}{2}$, 5, 10, 25 and 50 in. per second. In another model, 15 channels are simultaneously recorded side by side on the film.



Instruments Specialties Co.

Fig. 1—Deflection test of a diaphragm made from a constant modulus nickel alloy.

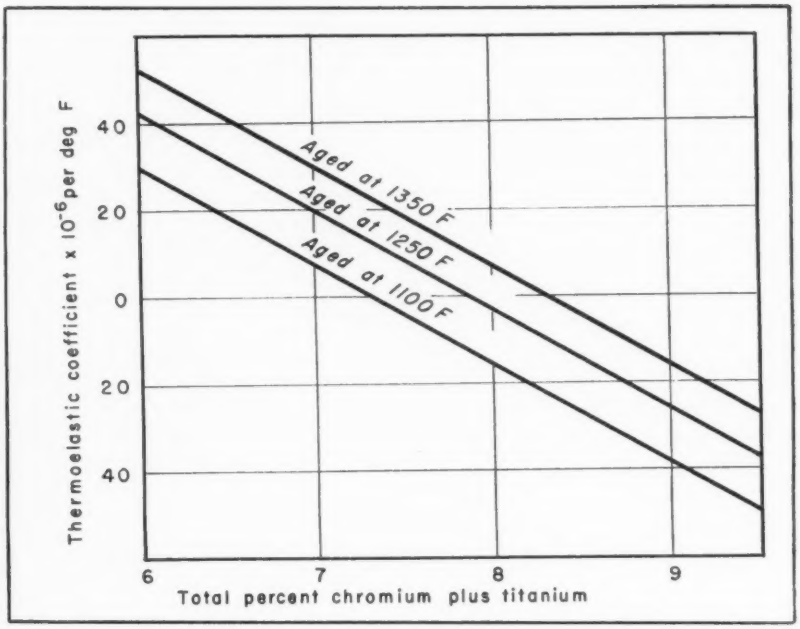


Fig. 2—Variation of thermoelastic coefficient with titanium and chromium additions and with age hardening temperature.

Age-Hardening Nickel Alloys

Properties and uses for a new group of nickel alloys having controlled elastic properties or controlled expansion characteristics.

FREDERICK C. OCHSNER
The H. A. Wilson Company

DURING THE PAST FEW YEARS a new group of nickel alloys has been developed in which the thermoelastic properties or the physical properties can be controlled within limits by varying the alloying elements and the heat-treatment. What distinguishes Ni-Span (registered trade mark of The International Nickel Company, Inc.) alloys is the fact that they not only have unusual thermoelastic properties, but are age hardening and can be heat-treated from the annealed condition or from any degree of cold working to obtain high physical and mechanical properties.

The combination of controlled elastic characteristics and high strength makes these alloys adaptable for use in springs, instrument parts, and other parts that must function accurately with changes in temperature.

Constant Modulus Alloy

There are five Ni-Span alloys, which fall into three groups: (1) High expansion alloy, (2) low expansion alloys, (3) constant modulus alloy. All these are made basically of nickel, iron and titanium, with or without some chromium, depending on alloy.

The outstanding alloy of the group is Ni-Span "C," which has a constant modulus of elasticity and modulus of rigidity over a useful range of temperature above and below ordinary room temperature (approx —50 F to 150 F). The combination of zero thermoelastic coefficient, which eliminates temperature change in the elastic moduli and the high elastic and strength properties obtainable through heat-treatment are important factors in the design of structural parts that require predicable elastic properties.

For certain applications, the alloy can be heat-treated to obtain a positive or a negative thermoelastic coefficient. If the thermoelastic coefficient is known, the corrected moduli of elasticity and rigidity can be gotten by using the following equations where

e or m = temperature coefficient
$d\theta$ = change in temperature in deg C from zero deg C
E and G = moduli of elasticity and rigidity respectively for alloy at zero deg C, psi
E' and G' = corrected values of elasticity and rigidity, psi

$$E' = E\,(1 - ed\theta) \quad (1)$$

$$G' = G\,(1 - md\theta) \quad (2)$$

Typical physical properties for this alloy are shown in Table I and in Fig. 2; the effects of alloying and age hardening on the thermoelastic coefficient are shown.

Uses for Constant Modulus Alloys

Ni-Span "C" is used in structural parts for instruments and devices where these parts must maintain constant elastic properties with change in temperature. Examples are tuning forks, diaphragms for pressure sensitive instruments, Bourdon tubes, strain gages and proving rings.

The alloy is used also for precision springs where a change in temperature must not affect the calibration of the spring, as in precision instruments and accurate scales. Usually some device to compensate for temperature error is used, such as bimetal thermostat elements, materials that change in electrical resistance or magnetic permeability or duplex spring assemblies made from two elements, one having a negative and the other a positive thermoelastic coefficient.

Ni-Span "C" frequently can replace these complex temperature compensating devices. For example, a Beryllium-copper spring under a given load

Table I—Properties of Constant Modulus Alloy

Nominal Composition: 42 Percent Nickel, 2.5 Titanium, 5.5 Chromium, 0.06 Carbon

	Solution Annealed	Heat-Treating Temperature			Solution Annealed Plus 50 Percent Cold Worked	Heat-Treating Temperature		
		1,100 F	1,250 F	1,350 F		1,100 F	1,250 F	1,350 F
Tensile strength, psi	90,000	150,000	180,000	175,000	135,000	185,000	200,000	200,000
Yield strength (0.2 percent offset), psi	35,000	95,000	115,000	115,000	130,000	160,000	180,000	180,000
Proportional limit, psi	15,000	70,000	65,000	65,000	55,000	105,000	110,000	105,000
Elongation in 2 inches (1 percent)	40	30	18	17	6	8	7	7
Rockwell hardness	78B	32C	37C	37C	29C	39C	42C	42C
Brinnel hardness	145	300	345	340	275	365	395	395
Modulus of elasticity $\times 10^6$, psi Modulus of rigidity $\times 10^6$, psi	24 10	27 10	26.5 10	26.5 10	25.5 10	27 10	27.5 10	27 10
Approximate thermo-elastic coefficient $\times 10^6$ per deg F	−15	−10	−5	0	−10	−5	0	+10
Thermal expansion coefficient $\times 10^{-6}$ per deg F (− 50 F to 150 F)	4.5	4.5	4.5	4.5	4.5	4.5	4.5	4.5
Thermal conductivity (Btu per sq ft per deg F per in.)	90	90	90	90	90	90	90	90
Electrical resistivity (ohms per sq mil ft at 68 F)	480 to 580*							
Electrical conductivity (at 68 F)	1.4 to 1.7 percent*							

* Depending on amount of cold-working and precipitation heat-treatment.

Table II—Mechanical Properties of High Expansion Alloy

	Solution Annealed	Age Hardened	50 Percent Cold Worked and Age Hardened
Tensile strength, psi	80,000	140,000	180,000
Yield strength 0.2 percent offset, psi	35,000	90,000	130,000
Proportional limit, psi	20,000	60,000	80,000
Elongation, percent	30	20	8
Modulus of elasticity $\times 10^6$, psi	22	24	24
Hardness Rockwell	74B	33C	40C
Nominal composition, percent	Ni—29, Ti—2.4, Cr—8.5, C—0.06		
Mean coefficient of expansion in./deg F	10.5×10^{-6} (72 F—1,000 F) 10×10^{-6} (72 F)		

will give an increase of deflection of approximately 1 percent for each 50 F change in temperature, so that over the temperature range of —50 F to 150 F there would be a 4 percent error in deflection. Over this same temperature range a Ni-Span "C" spring will show no appreciable change in deflection (less than .2 percent) while a temperature change of over 500 F is required to give a 4 percent increase in deflection. This alloy has other unusual spring properties, such that after heat-treatment, the hysteresis or back error is less than 0.04 percent and the error due to creep under safe working loads is less than 0.02 percent.

High Expansion Alloys

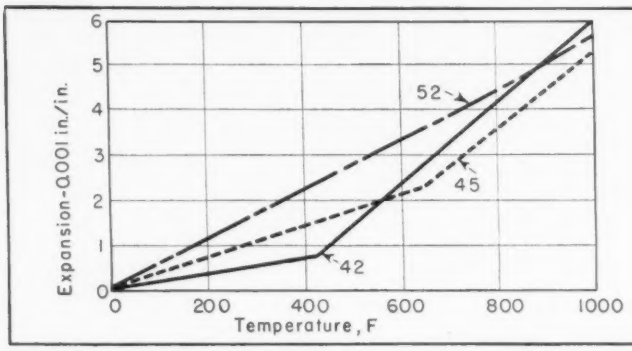
Ni-Span Hi is a precipitation hardening alloy with a coefficient of expansion near the maximum possible with an austenitic alloy, in addition to its high physical properties. The advantages of this type of alloy are its high strength, resistance to thermal and mechanical shock and improved elevated temperature properties over alloys of similar compositions. It is used for special springs, where high expansion and high strength are required, and in other members in instruments and control equipment.

Low Expansion Alloys

Ni-Span Lo is a precipitation hardening alloy having an expansivity slightly greater than invar and, in the cold worked and age hardened condi-

Table III—Mechanical Properties of Low Expansion Alloys

	Ni-Span—LO-42			Ni-Span—LO-45			Ni-Span—LO-52		
Composition	Ni—42 Percent Ti—2.4 Percent	C—0.06 Percent Bal—Fe		Ni—45 Percent Ti—2.4 Percent	C—0.06 Percent Bal—Fe		Ni—52 Percent Ti—2.4 Percent	C—0.06 Percent Bal—Fe	
Condition	Annealed	Hardened	50 Percent C. W. and Hardened	Annealed	Hardened	50 Percent C. W. and Hardened	Annealed	Hardened	50 Percent C. W. and Hardened
Proportional limit, psi	20,000	65,000	110,000	19,000	61,000	103,000	15,000	55,000	100,000
Yield strength offset, psi	40,000	120,000	165,000	37,000	113,000	155,000	35,000	95,000	140,000
Tensile strength, psi	90,000	165,000	195,000	85,000	155,000	183,000	85,000	120,000	175,000
Elongation, percent	32	14	5	30	13	5	27	17	5
Hardness Rockwell	78B	34C	40C	74B	33C	37C	71B	32C	36C
Modulus of elasticity $\times 10^6$, psi	21.0	22.0	22.5	21.0	22.0	22.5	21.0	22.0	22.3



COEFF. OF EXPANSION X 10^6/DEG F

Alloy	Min coeff.	Max coeff.	Mean coeff. (0-1000 F)	Inflection temp, F
42	1.8	8.8	5.8	430
45	3.4	8.6	5.2	650
52	5.3	6.5	5.5	8.20

Fig. 3—**Expansion characteristics and inflection temperatures of low expansion alloys containing three different amounts of nickel.**

tion, much higher physical properties. The nickel content of this alloy is 42, 45, or 52 percent. In Fig. 3, the expansivity of these compositions is shown. These alloys are useful in instruments and other equipment where high strength and low expansion coefficients are essential. Examples are special springs, instrument frames and precision machine parts.

Ni-Span—LO-42 has been used in arbors for carboloy faced rolls in rolling mills because it has an expansion rate close to that of cemented tungsten carbide, and high physical properties. (See Table III.) This alloy is also used as a backing material for tungsten carbide tipped tools.

Directory of Commodity Specifications Expands

A LIST OF NEW STANDARDS and specifications in general use for commodities produced in this country has been collected recently by the National Bureau of Standards to supplement its 1945 National Directory of Commodity Specifications. This 322 page supplement and the 1,300 page Directory include not only specifications formulated by government purchasing agents but also those of more than 500 non-government organizations having national recognition.

Each standard, specification, simplification, code, safety rule and similar document for general use was examined by NBS and arranged to give the user as complete information as possible.

The Federal Government, as the largest single purchaser of consumer goods in the country, has made its purchases under carefully prepared specifications. Prior to 1921, each agency of the government had its own specifications. To avoid duplication of effort, the Federal Specifications Board was established and given the duty of compiling and adopting these standard specifications. The Board is headed by the Director of NBS, E. U. Condon is the present director, and consists of qualified representatives from various branches of the government.

Federal specifications, of which there are now more than 1,900, are first formulated by one of nearly 80 Federal Specification Technical Committees, consisting of official representatives of the various departments and establishments interested in the commodity. Members of the staff of NBS act as chairmen of about 40 percent of these committees, with advice from commercial and industrial concerns. Specifications prepared by the Technical Committees are submitted to all government agencies and interested engineering and technical societies for criticism before it is placed in final form.

The National Directory of Commodities Specifications (Bureau of Standards miscellaneous publication M-178) and its supplement can be obtained from the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C. The Directory costs $4 and the supplement $2.25. All orders should be accompanied by check or money order made payable to the Treasurer of the United States.

PLAIN SLEEVE BEARINGS

Materials and Design

Engine Bearings

Iron, Steel, Porous Metal, Wood, and Plastic Bearings

Carbon-graphite and Rubber Bearings

Hydrodynamic Bearings for Large Stationary Machinery

Results of a study and survey made by the editorial staff of Product Engineering. All engineering data appearing in this article are from authorities on bearing theory and design, bearing manufacturers and users, and bearing literature, published and unpublished.

Factors that affect the load capacity and life of bearings . . .

1 APPLIED LOAD . . .

(a) Magnitude

(b) Type — constant, varying uniformly with time, varying nonuniformly with time, or shock

(c) Direction — unidirectional or varying in direction with time

2 RUBBING SPEED . . .

(a) Magnitude

(b) Type — constant, oscillating

3 LUBRICANT . . .

(a) Viscosity

(b) Oil or lubricant "in" and "out" temperature

(c) Pressure

(d) Flow

(e) Oiliness

4 DIMENSIONS . . .

(a) Clearance-diameter ratio

(b) Length-diameter ratio

(c) Bearing thickness (liner plus back)

(d) Bearing liner thickness

(e) Bearing and shaft finish and hardness

5 MATERIALS . . .

(a) Bearing materials

(b) Shaft materials

6 OTHER FACTORS . . .

(a) Shaft stiffness

(b) Corrosive agents in lubricant or atmosphere

(c) Dirt in lubricant or in bearing

(d) Bearing housing stiffness

(e) Contact between bearing and housing

(f) Degree of lubrication when starting or stopping

Bearing Performance Factors

Design formulas for full journal bearings derived from basic hydrodynamic and thermodynamic theory assume

1. Load is uniform and unidirectional
2. Speed is constant and does not change direction
3. Bearing dimensions, once established, do not change
4. Bearing and shaft are absolutely smooth and circular
5. Shaft has infinite stiffness
6. No dirt or metal in lubricant or between bearing and journal
7. Lubricant is perfectly distributed to all parts of the bearing surface
8. Bearing is infinitely rigid
9. Oil film is never broken
10. Oil temperature in bearing is known

Obviously few bearing installations entirely satisfy these assumptions. Actually few bearings, laboratory tested under controlled conditions, justify these assumptions although they approach them. But such laboratory tests do have great value because they allow the designer to qualitatively compare bearing materials.

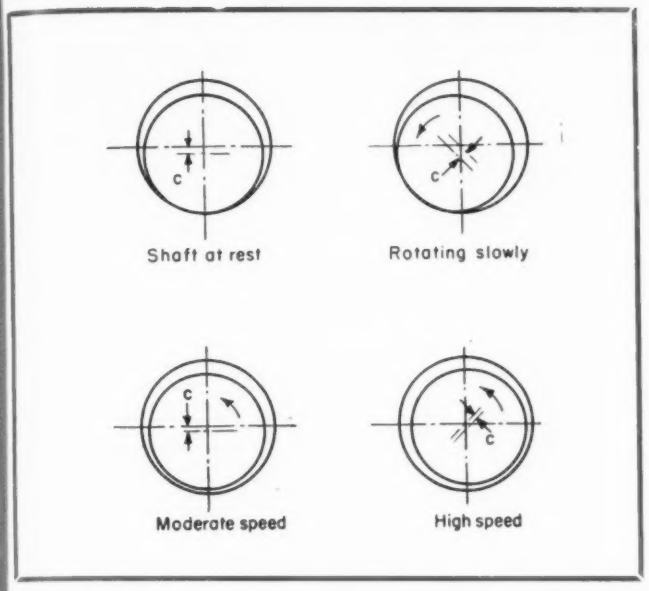


Fig. 1—Formation of supporting film in plain sleeve bearings. Eccentric journal and bearing pump oil around clearance space.

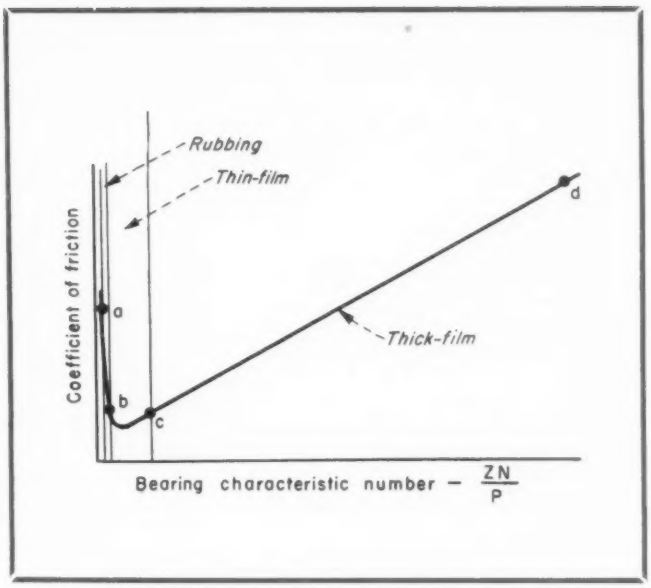


Fig. 2—Relation between coefficient of friction and bearing characteristic number ZN/p. Most bearings operate near point C.

Design Concepts

THE APPARENT PAUCITY of application design data for all plain sleeve bearings arises not from engineering neglect of this subject but to the difficulty of applying theory in practice. While the design of plain sleeve bearings that operate with a full fluid film follows the classical engineering approach associated with many other machine or structural elements, the design of bearings operating with partial solid-to-solid contact involves little, if any, analytical approach.

For most bearing installations, bearing design is a mixture of hydrodynamic and thermodynamic theory, material properties, lubricant properties, test data, and the designer's engineering knowhow—the specific design approach being determined by bearing type and use.

The basic concept of a plain sleeve bearing assumes a journal (bearing portion of shaft) rotating inside a supporting bearing but separated from it by a film of lubricant. As long as the film is not broken, the performance of the bearing can be determined from hydrodynamic theory. But if the film is broken, bearing performance becomes a function of a number of complex operating variables that cannot, as yet, be formulated into a simple theory. Because of this there has been a feeling among many designers that plain bearings cannot be designed on paper. However, there does exist much empirical data on many types of plain bearings. These data do bridge the gap, to some extent, between theory and practice, and if used with discretion, can be of assistance in taking some of the guesswork out of design.

ZN/p AND FILM THICKNESS

Bearings are classified as thin-film lubricated or thick-film lubricated. Fig. 1 shows four positions for a bearing with a vertical down load, operating at different speeds. As journal speed is increased, more oil is dragged around and forced through the region of minimum clearance between the bearing and journal. The hydraulic effect of this action is to increase positive pressure in the oil film in this region. The net fluid pressure acting on the journal and on the bearing, forcing them apart, determines the thick-film load capacity of the bearing.

Oil film thickness for any one bearing is a function of several operating variables, but is usually expressed in terms of a bearing characteristic number ZN/p, where Z is the absolute viscosity of the oil in the bearing, in centipoise, N is the journal speed in rpm, and p is the unit bearing load in psi (arbitrarily expressed as the load divided by bearing diameter times length).

If lubricant viscosity or speed goes down or load goes up, the characteristic number gets smaller; and the oil film thickness is reduced. This relation is shown in Fig. 2. Here ZN/p is plotted against coefficient of friction. Starting at the right side of the curve, if ZN/p is reduced, friction drops. But at some small value of ZN/p, friction increases rapidly with further decrease in ZN/p. The rapid increase in friction comes about when the oil film has been ruptured. Bearings are generally not designed to operate in region *a-b* and may fail if they do operate in this region for any length of time.

It would appear that the best procedure is to design bearings to operate in region *c-d*, but because power losses in this region may be high and lubricating problems severe, most bearings operate in the region *b-c* or slightly to the right of it.

In this region of lubrication, oil film thickness may be as little as 0.001 in. Obviously dirt in the lubricant, small changes in lubricant viscosity, and shaft deflection have the effect of further reducing oil film thickness.

If the oil film is ruptured periodically or intermittently, and bearing and journal physically contact each other, bearing performance depends on the bearing and journal materials and their action on each other. Engine bearings, in particular, must be designed to anticipate this condition.

Bearing Types

BEARING TYPES are listed in the accompanying table, together with other data that defines these types. Note that bushings and sleeve bearings are both bearing types. Bushings perform the same function as sleeve bearings but are made of the same material throughout. The exceptions are sintered powdered metal bushings referred to as bearings. Sleeve bearings may be full round or split, with a lining of low friction material applied to a back or shell of steel or bronze.

It is significant that each type requires a different design approach. In the following pages these types will be discussed, starting with empirically designed bearings and ending with hydrodynamic bearings for turbines and other large, stationary machines—bearings that lend themselves to a more rigorous analysis.

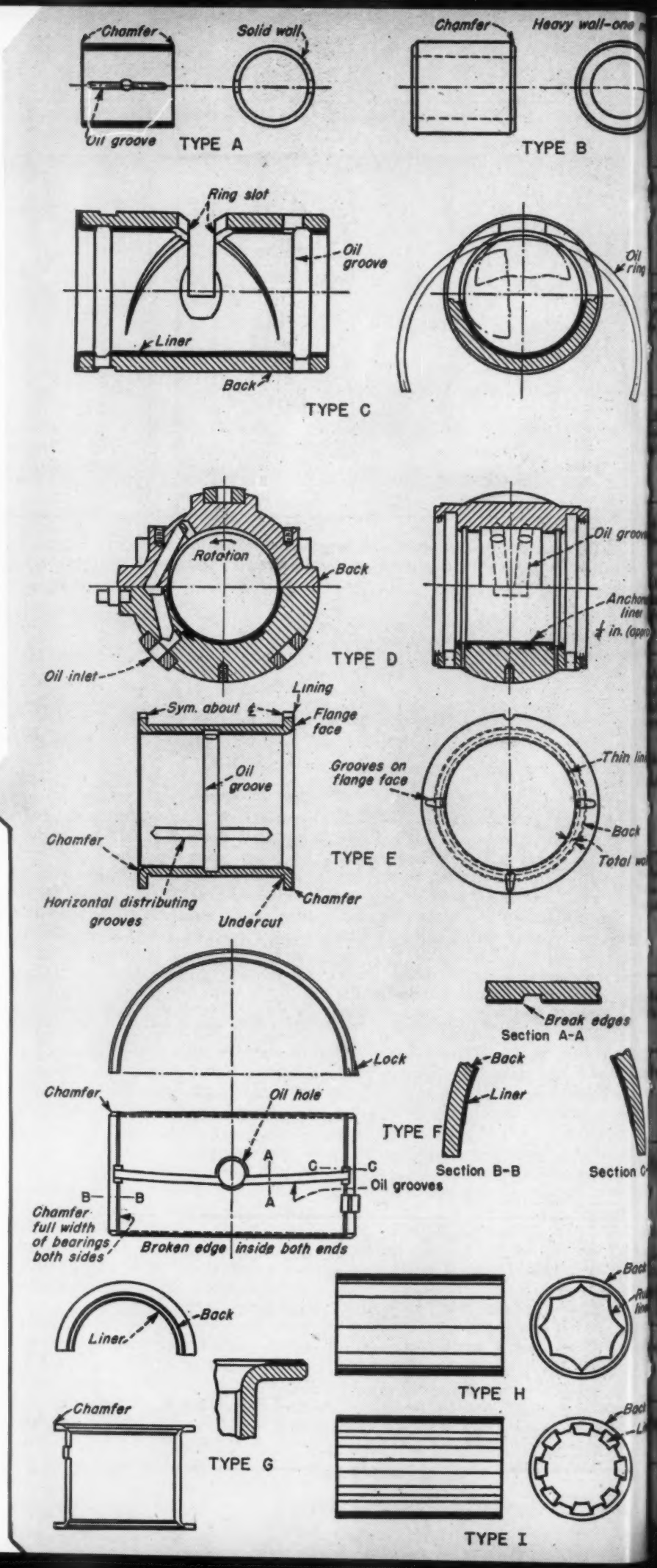
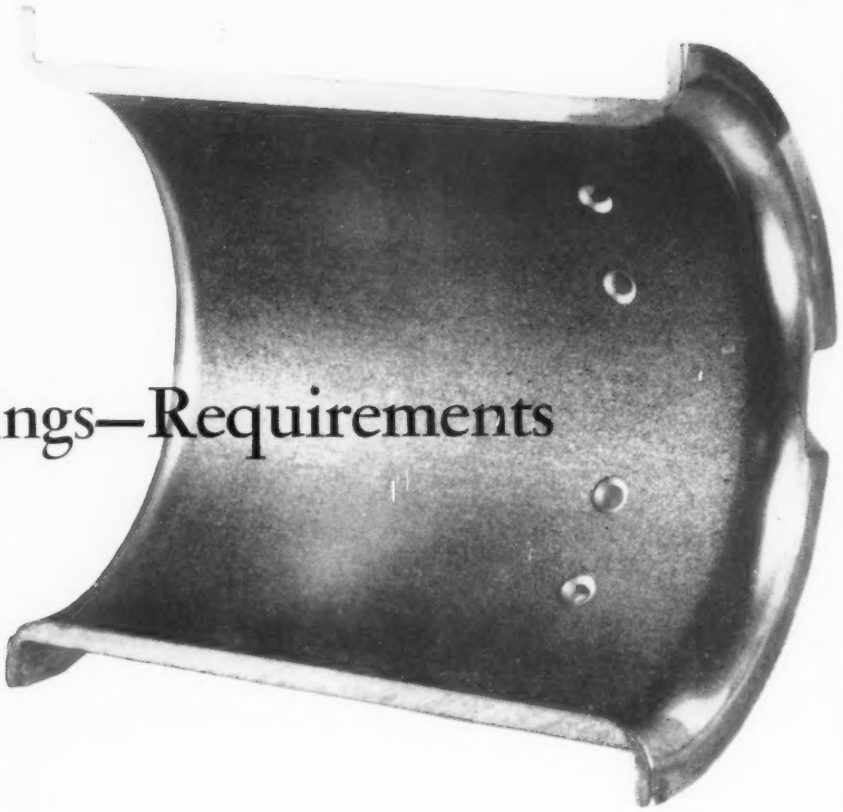


Table I—Types of Sleeve Bearings

Type	Lubrication	Shape	Predominant Materials	How Made	Characteristics	Operating Limitations	Typical Uses
ning	Thin-film	A	Bronze	Formed sheet, cast, machined from bar stock, or made from tubing	Low cost, simple construction, moderate loads. Used for operation under adverse lubrication conditions	Not good for shaft speeds over 800–1000 fpm and loads over 800 psi	All types of machinery and equipment
us ring (oil-	Thin-film	B	Bronze, iron, aluminum	Powdered metal, compacted and sintered	Low cost, self lubricating, quiet	Usual speed range: 75 to 1500 fpm. Usual bearing pressure limit —100 psi. Not good for heavy duty, general purpose use	Fractional hp motor, accessory drives, consumer products, controls
round- d (heavy g)	Thick-film	C	Babbitt liner, steel or cast iron back	Liner centrifugally cast to back, or liner poured around mandrel, centered in bearing back	Long life, low friction. Used only with good lubrication	Not good for pressures over 300 psi. Used principally for steady load applications	Large electric motors, turbines, heavy transmission shafting
-lined vy lining)	Thick-film	D	Same as bearing type C	Same as bearing type C	Same as bearing type C	Same as bearing type C	Same as bearing type C
round- d (light g)	Thick-film	E	Steel or bronze back with liner of one or more of several bearing materials	Liner centrifugally cast in tube, liner electroplated on back	Heavy duty general purpose—good for dynamic loads	Maximum practical limits—5000 psi and 4000 fpm	Gas, diesel engines, compressors, heavy duty machinery
t-lined nt lining) n and ged	Thick-film	F and G	Same as bearing type E	Same as bearing type E or by bonding liner to back in strip form and then forming to half round	Same as bearing type E	Same as bearing type E	Same as bearing type E
ber	Thick-film	H	Rubber liner and cast iron, brass or steel back	Molded liner bonded to back	Low friction, high resistance to abrasion, shock absorbent, long life	Not generally used at more than 80 psi. Require adequate water lubrication	Marine propeller bearings, rudder pintles, pumps, turbines
lded stic inate	Thick-film	I	Cotton and phenolic resin	Cotton impregnated with phenolic resin and molded	Low friction, high shock resistance. Stronger than babbitt lined bearings when used with water lubrication	Must be well cooled. Has low conductivity. Max. speed—400 fpm. Max. pressure—2500 psi (average)	Mill roll necks, pumps, propellers
lded stic	Thick-film	B	Phenolic	Molded to shape	Low friction, clean	Must be well cooled. Limited to low loads if run at high speeds	Dairy, textile, baking, food machinery
od	Thick-film	B	Lignum vitae	Turned	High shock resistance, high load capacity, at medium speeds if water lubricated	Not over 2000 psi—(water lubricated). Not over 200 F	Mill roll necks
od	Thin-film	B	Lignum vitae, maple, etc.	Turned and oil impregnated (if needed)	Self lubricating, clean, long life, low cost. Used principally for shaft guides	Light loads if speeds are high. Not over 150 deg F	Conveyors, textile, food machinery
bon- phite	*	B		Machined from sintered bar stock or molded and sintered	Light duty use. No lubrication required	Not over 850 deg. Up to 600 psi at low speeds. Up to 20,000 rpm at low pressure	Electric motors, meters, conveyors
t iron	Thick-film	B	Cast iron	Machined from cast stock	Low friction, low cost, develops high glaze in service	Not over 500 psi and 130 fpm	Cam shafts, link pivots, light transmissions
el	Thick-film	B	Steel	Machined from wrought stock	Low cost	Light loads; speeds not over 150 fpm	Guide bush

*No fluid lubricant.

Engine Bearings—Requirements



PLAIN SLEEVE BEARINGS, used in engines for automotive and aircraft use, diesels, compressors and other types of reciprocating machinery must take heavy dynamic loads at high speed. The trend to compactness, light weight, higher power output and long life in these machines has accelerated work in developing bearings with higher load ratings and longer life.

Through experience, experiments, research and performance, criteria for engine bearings have been set up. The degree to which any bearing satisfies these requirements establishes its quality. These requirements are:

FATIGUE RESISTANCE

Ability of bearing material to resist cracking under repeated stress. Fatigue is caused primarily by the direct effect of applied alternating load. The repeated stresses produced affect the whole bearing structure including the liner bond to the backing.

CONFORMABILITY

Tendency of bearing material to flow or creep slightly to allow journal and shaft to conform to each other. This allows for small mechanical and physical inequalities at assembly without affecting bearing performance later.

EMBEDDABILITY

Ability of bearing material to absorb or embed dirt and grit within itself. Metal particles or dirt not washed out by the lubricant destroy the oil film in places. If these particles can enter the body of the bearing material, they are out of the way and can do no harm. The softer metals naturally have better embeddability characteristics.

ANTI-SEIZING

Ability of bearing material to avoid galling, seizing or loading up under metal-to-metal contact. Probably the property of a bearing material to hold a film of oil under severe operating conditions, preventing point-to-point welding between bearing and journal.

CORROSION-RESISTANCE

Ability to resist attack by lubricant or agent contained in lubricant.

Capability of operating with unhardened journals. The harder bearing materials require hard journals even if journal and bearing surfaces are smooth.

BONDABILITY

Ability of bearing material to bond to backing material. Strength of bond when subjected to fatigue and thermal stresses.

TEMPERATURE-STRENGTH RATIO

Strength of bearing material at operating temperatures compared to strength at room temperatures.

THERMAL CONDUCTIVITY

Ability of bearing material to absorb and conduct heat generated by bearing friction. Lower operating temperatures improve compressive and bond strengths.

From past records of automotive bearing failures, bearing quality requirements are weighted with regard to their relative performance as follows:

Fatigue strength	60 percent
Conformability	10
Embeddability	10
Anti-scoring	5
Corrosion resistance	5
Shaft hardness	5
Bond	2
Temperature-strength ratio	2
Thermal conductivity	1
	100 percent

There are two stumbling blocks in the way of selecting bearing materials to fit these requirements. First, most bearing characteristics cannot be measured and given finite values useful for all bearing applications. Second, no bearing material satisfies all these requirements.

Even though automotive engine manufacturers have spent much time in an effort to determine rational methods for bearing design, they maintain that bearing performance cannot be accurately determined in advance. Performance in service is the only criterion of performance.

However, by choosing materials having bearing properties that come closest to fulfilling the requirements set by operating conditions, and by relying on current bearing usage, the designer removes some of the factors of ignorance inherent in engine bearing design.

Engine Bearings—Materials

SOME OF THE MORE COMMON bearing materials are:

Tin-base babbitts
High-lead babbitts
Cadmium alloys
Copper leads
Copper leads—overplated
Bronzes
Aluminum alloys
Silver
Silver-overplated

With the exception of the bronzes and some aluminum bearings, most of the materials in this list are not used by themselves but as liners in bearings having steel or bronze backs. This is done to improve mechanical strength and occasionally to reduce cost. The bronzes and aluminum alloys are used alone because their physical properties approach those for low carbon steels and they need no support if loads and speeds are not too high.

TIN-BASE BABBITT bearings have been used for more than a hundred years. More tin-base alloys are used today than any other material with the exceptions of the high-lead alloys. Babbitt bearings are made by lining fluxed steel or bronze tubes with cast babbitt and then machining; or by bonding babbitt to a steel or bronze strip in a continuous mill and then forming the bonded strip to shape.

There are many tin-base alloys or mixtures, all having about the same properties. The fatigue life of all is nearly the same. But the main reason for the popularity of babbitt is its excellent anti-score properties. It is least sensitive to oil-film failure and metal-to-metal contact from shaft distortion.

Babbitt is easily bonded and cast. Added advantages are its conformability, which allows good run-in, embeddability, allowing dirt to be absorbed without scratching the shaft, and resistance to corrosive acids.

The disadvantage in the use of tin-base babbitt for bearings, is its low fatigue resistance compared to other materials. Also at comparatively low temperatures, the hardness and strength of babbitt drops appreciably. Fig. 1 shows its hardness compared to other materials in a range of temperatures.

LEAD-BASE BABBITT bearings are similar to tin-base babbitt bearings in all characteristics. In the past, when liners were thicker than now, the lead-base babbitts were not as good. But in precision bearings having a thin layer of the bearing material on a steel or bronze back, their fatigue strength is sometimes greater than tin-base babbitts. The shortage of tin during the war accelerated the use of this material.

One distinct lead-base alloy is the hardened lead alloy; hardened with 1 to 2 percent tin, calcium and small percentages of other alloys, instead of tin and antimony. It is not as resistant to corrosion as the softer lead-babbitts.

CADMIUM BASE bearings have longer fatigue life than babbitt bearings and about three times the endurance in running time. They also can replace babbitt bearings because clearances need not be changed. Cadmium base bearings are easily corroded unless 0.5 to 0.75 percent of indium is plated on and diffused into the finished bearing. Keeping bearing temperatures below 250 F also retards corrosion. Cadmium base bearings are not as good as

Table I—Comparative Properties of Several Metals Used for Liners in Engine Bearings

Bearing Property	Bearing Material							
	Tin-Base Babbitt	High Lead Babbitt	Cadmium	Copper-Lead	Copper-Lead Overplated	Aluminum	Silver	Silver Overplated
Fatigue resistance	8	12	32	47	47	80	100	100
Conformability	83	83	58	13	100	53	2	100
Embeddability	100	75	58	37	50	18	7	12
Anti-scoring	100	93	56	38	75	86	6	75
Corrosion resistance	100	75	37	25	83	100	100	100
Shaft hardness	100	100	85	38	100	38	2	12
Bondability	80	60	90	100	100	...	100	100
Temperature-strength ratio	6	8	10	16	16	40	100	100
Thermal conductivity	16	8	55	70	66	50	100	96

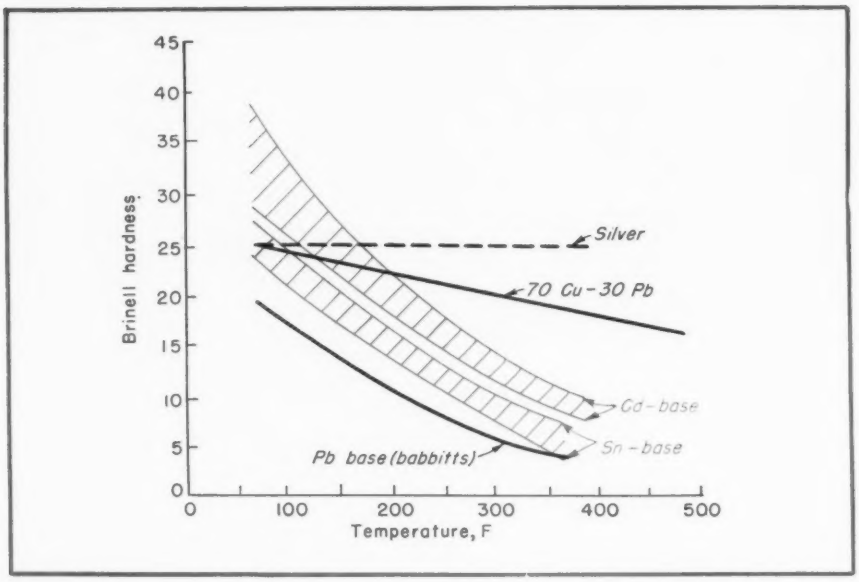


Fig. 1. Hardness variation of babbitts for normal operating range.

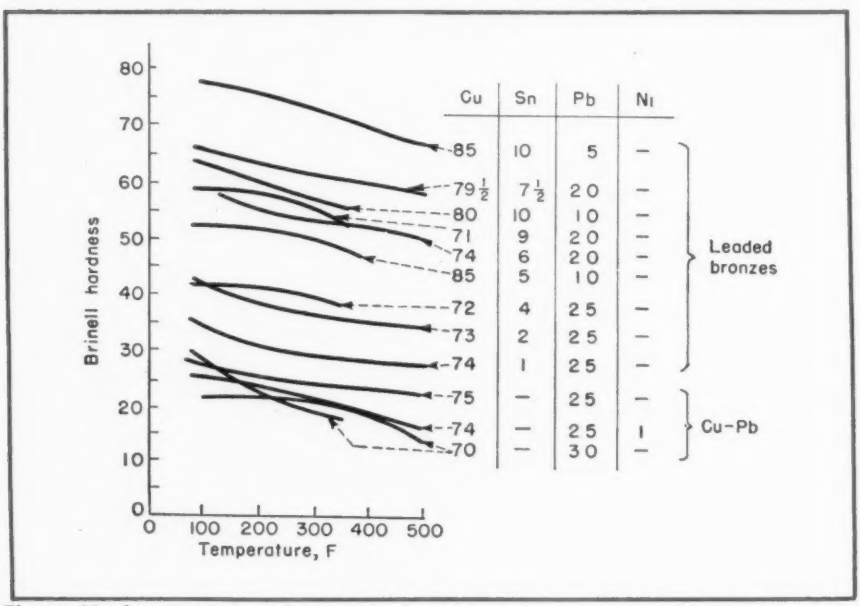


Fig. 2—Hardness variation of copper-leads and leaded-bronzes normal operating range.

babbitt bearings in conformability, embeddability and score resistance. Cadmium is more difficult to bond than babbitt. Both high cost and availability of better bearing materials have limited the use of this material.

COPPER-LEAD bearings are made from mixtures of copper and lead, usually 25 to 45 percent lead often with small amounts of other elements. The copper provides mechanical strength and fatigue resistance, and the lead anti-score properties. Copper alloy bearings give four to five times the life of babbitts and under high load at high temperatures, are even better. They score more readily than babbitts.

OVERPLATED COPPER ALLOY bearings are made with a thin overlay of lead alloy to improve anti-scoring, conformability, embeddability and corrosion resistance properties. The softer overlay backed by the hard copper matrix results in a better bearing than the plain copper-lead bearing. Without the lead alloy, the soft lead in the copper matrix may be dissolved by the fatty acids normally present in oxidized mineral oils.

Copper-lead bearings are relatively expensive if manufacturing specifications are rigid. Difficulty in bonding and prevention of segregation are two factors adding to the cost.

LEADED-BRONZE AND TIN-BRONZE. Leaded bronzes are mixtures of bronze and up to 25 percent lead. The higher tin, lower lead, alloys are harder (up to 80 Brinell). Because of their hardness, they usually are used in bushings for shafts running at moderate speeds and loads; the hardness of the alloy being an inverse measure of the speed at which they can be run.

Tin-bronzes contain approximately 10 percent tin. They are hard (up to

Table II—Definitions for Copper and Bronze Plain Sleeve Bearing Materials

Bearing Material	Principal Metals	Remarks
Copper-base alloy	Copper-lead-tin in varying proportions ranging from soft alloys of copper and lead to hard alloys of copper and tin. Soft alloys usually bonded to steel back. Hard alloys used in bushings.	Restricted to slower speed applications where utmost in bearing properties are not needed.
Copper-lead alloys	25 to 45 percent lead with possible addition of small amount of silver. Alloys are soft and are usually used with steel back.	Difficult to bond to steel.
Leaded-bronze	5 to 25 percent lead and 3 to 10 percent tin. Zinc often used as replacement for tin.	Generally used as cast bushings where speeds and loads are not too high. The higher the tin content, the lower the allowable speeds.
Tin-bronze	Up to 10 percent tin, small amounts of lead. High tin alloys are very hard. High strength but only fair bearing properties.	Used only for applications where speed is low.

170 Brinell) and cannot be used for high-load, high-speed applications. Leaded bronzes and tin-bronzes are used in bushings for all types of machinery. In Fig. 2 is shown the relative hardness of a number of copper-lead and leaded-bronzes.

ALUMINUM BEARINGS. Alloyed with small amounts of tin, nickel, copper, silicon or cadmium, aluminum bearings show up well, although embeddability is low. Journals must be hard, but even with hardened journals, aluminum bearings occasionally seize to the point of permanently welding thin sections to the journal. Aside from seizing, aluminum bearings have good fatigue and yield strength when backed by steel. Bonding of aluminum to steel has not always been satisfactorily done in production, a factor limiting its use.

Another objection to aluminum bearings is their high coefficient of expansion compared to steel. Installed bearings have a tendency to loosen in service. Work hardening the bearings before installation reduces this tendency.

Aluminum has good thermal conductivity and excellent resistance to corrosion. Solid bearings are not quite as strong as copper-lead bearings, but steel back aluminum bearings are equal to silver. The use of aluminum bearings is not widespread, but with improvements in the manufacturing process, they probably will be used for many applications requiring a high load capacity material, reasonably priced.

SILVER is the best of all bearing materials from the standpoint of fatigue resistance, but is hard and does not have the anti-friction properties of babbitt. Bearing properties are improved by overplating with lead and indium, lead and tin, or by gridding the silver and filling with lead alloy or lead and indium.

Most silver bearings are made of a steel back electroplated with silver. Silver has a good record of performance and has become the standard, for the present at least, for the ultimate in bearing materials. Overplated with lead-indium, silver bearings show up well in anti-seizing. Embeddability is not as good as the softer materials, so silver bearings are more easily damaged by dirt and grit.

In Table I, the foregoing materials

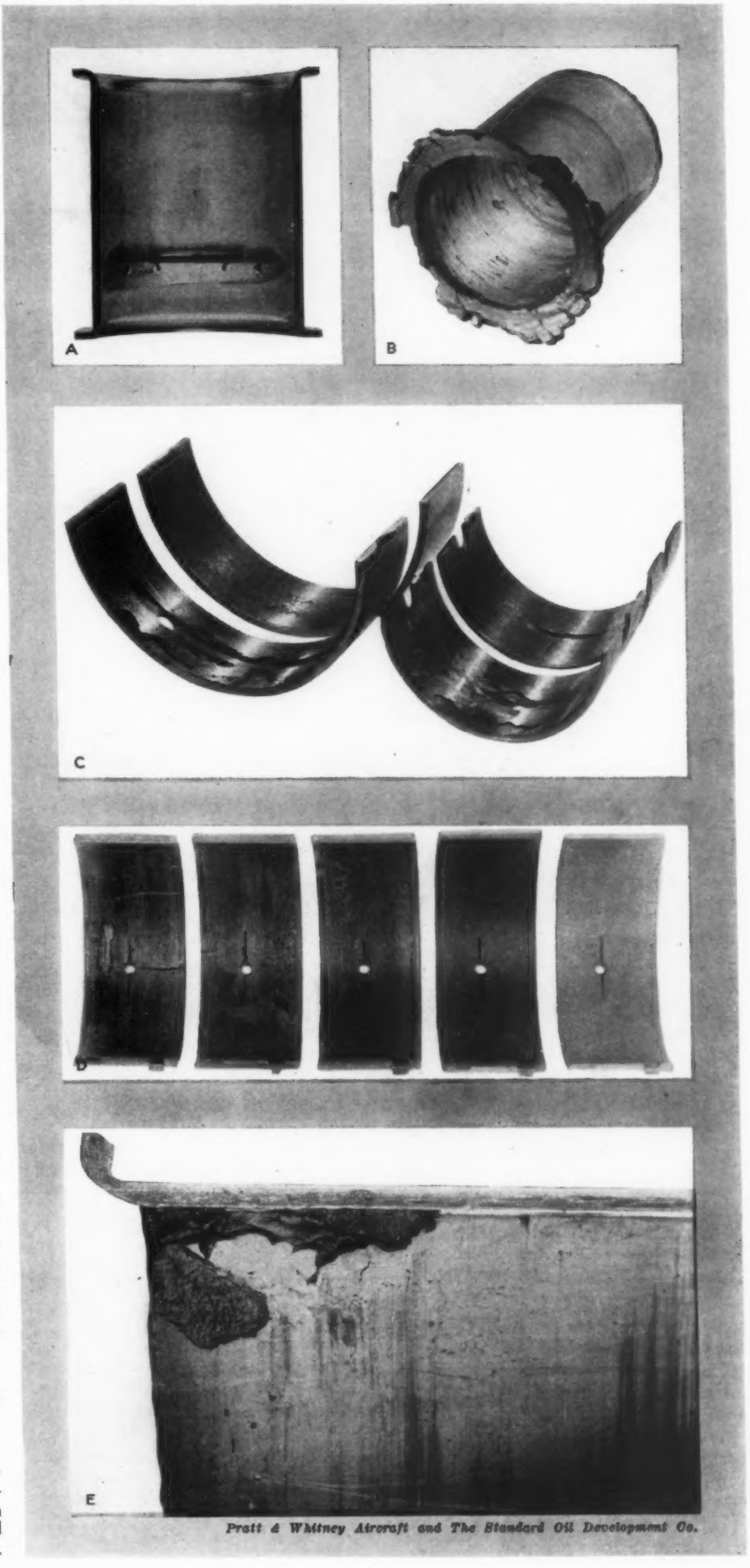


Fig. 3—Typical failures: (A) Silver bearing—seized (B) Aluminum bearing—seized (C) Cadmium bearing—corrosion (D) Copper-lead—scored, corroded and mechanical failure (E) Babbitt—fatigue. *Pratt & Whitney Aircraft and The Standard Oil Development Co.*

Table III—Specifications for Bearing Metals

Metal	Specification					
	SAE	AMS	ASTM	Federal	U. S. Navy	Other
Aluminum		4082 4150				
Babbitt-lead base	13	B23 *9*				
	14	B27 *7*		QQ–M–161 *7*		
	15	EA–B23a				
Babbitt-tin base	10	4800	B10, B23–26 *1* *1*	QQ–M–161 *1*		
	11 12		B23 *2*	QQ–M–161 *2*	46–M–2 *2*	
Bronze		4625				
	43	4860	B54–27	QQ–B–726	49–B–3	
	62	4845	B143–44T *1A*	QQ–B–691 *5*	46–M–6	
		4846				
		4862		QQ–B–726 *c*	49–B–29 *A*	
		4870				
		4871				AN–QQ–B–672
		4872				
		4873				AN–QQ–B–672
	791 795					
Bronze (porous)		4805				
Bronze-leaded	40	4855	B145–44T *4A*	QQ–B–691 *2*		Air Corps 11306–A
	64	4842				
		4822				
		4825				
		4840	B66–28		46B22 *V*	
		4827				
	792 793 794					
Cadmium	18 180					
Copper lead	48 480	4820				
Copper base alloys	65 65+Ni 640 63 620 621 622 660 66					
Silver	17P 17C 17R	4815 4817				

are rated in the order of their performance. These ratings vary, of course, with the choice of alloys or mixtures, and with specific bearing applications.

Although silver has been selected as the ultimate in bearing materials and is given a rating of 100 for several of the bearing properties listed, future research may produce another metal or metal alloy with better characteristics.

Engine Bearings — Material Specifications

There are hundreds of specifications for engine bearing materials. Each bearing manufacturer has his own, which may or may not be the same as standard bearing material specifications. Many of these are intended for bearings designed for special purposes. However, most materials are made to the specifications listed in Table III. The materials listed in the table are used either for bushings or for bearing liners. The copper base alloys listed are used principally for bushings. The silver, babbitt and aluminum alloys listed are used principally for bearing liners. The bronze-leaded alloys listed are used for wrist-pin bearings and other heavy load applications.

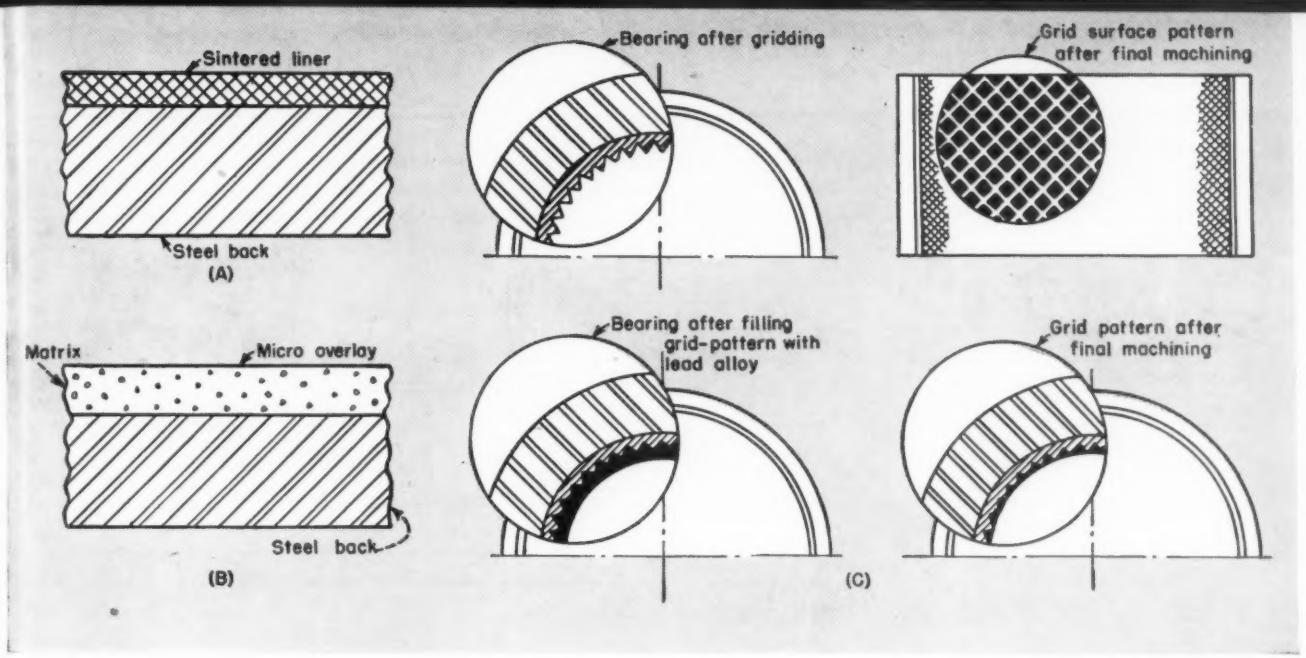


Fig. 1—Common constructions for engine bearings used to improve bearing properties of hard materials. (A) Sintered liner (B) Overlay (C) Gridded surface. All three constructions are means for obtaining strong bearings with good wearing qualities.

Engine Bearings—Current Design

Cost

Frequently, bearing cost is a design factor. The designer has some control over this item provided bearing operating conditions are not too severe and allow a choice of materials. Finished engine bearings show the following order of increasing cost.

Bronze and aluminum bushings
Lead-base babbitts
Tin-base babbitts
Cadmium alloys
Copper-leads
Silver

Construction

Bearing construction determines performance as much as the materials from which the bearing is made. This applies to both the metallurgical and physical construction of the bearing. For heavy-duty, high-speed engine bearings, a number of methods are used to improve their properties.

Most of these bearings are made with a liner of one or more common bearing materials bonded to a steel or bronze back. The back may be used for one or more reasons. It may provide the strength lacking in the bearing material, better heat conductivity, or save cost by limiting the amount of bearing material needed. The choice as to whether the back is made of steel or bronze depends on several factors. The advantages and disadvantages of bronze compared to steel are

Advantages

BRONZE BACK

1. Greater heat conductivity
2. Coefficient of expansion near babbitts
3. Nonscoring of shaft if liner fails (and oil supply does not fail)

Disadvantages

1. Cannot be bonded to cadmium, silver, or copper-lead
2. Lower fatigue resistance
3. Higher cost
4. Lower compressive strength

Table I—Relative Values of Dry Coefficient of Friction*

Bearing Material Rubbing on Steel	Coefficient of Friction
Steel	1.00
Copper	0.90
Copper film on steel	0.30
Lead	1.20
Lead film on steel	0.18
Copper-lead alloy	0.17
Indium	2.00
Indium on steel	0.08
Silver	0.55
Indium film on silver	0.10

* Values shown are useful for comparative purposes only. Steel on steel is taken as 1.00

The way the back is made also affects bearing performance. Bronze backs usually are centrifugally cast rather than sand cast in order to get sound structure. Steel backs are made from strip or tubing.

Aside from backing material, bearing liner construction does much to determine performance. High fatigue-strength bearing materials (see table page 135) are poor in conformability and embeddability. Several constructions are used to take advantage of the high fatigue strength of these materials and the good conformability and embeddability of the softer materials.

Overlays

An overlay is defined as a thin layer (about 0.001 to 0.005 in.) of lead, babbitt, tin, or indium, over the normal bearing material of silver, copper-lead, leaded-bronze or other hard material. Bearing strength is reduced only slightly and the soft overlay allows for absorption of dirt and adjustment of bearing to journal.

Mixtures of Hard and Soft Materials

A mixture of a hard material like a copper alloy and a soft material like lead alloy is sintered and bonded to the bearing back. The hard copper

Table II—Allowable Pressures for Engine Main and Connecting Rod Bearings

Bearing Material	Pressures at failure for bearings tested under ideal conditions			Recommended for design				Actual values from P.E. survey
	1	2	3	4	5	6	7	8
Tin base babbitt	1750			800–1500	1000–1500	2500*	1000–1200	—
Tin base (micro)	2700			2000–4000			1200–1500	—
Lead base babbitt	2200			800–1500	1800	1800*	1000–1200	2130
Lead base (micro)	2700			2000–4000			1200–1500	2400
Cadmium alloy	—			1200–1500	1800–3850		2500	1675
Copper-lead (overplated)	3600 7500	8300		1500–2500 3000–4000	1800+		1500–3800 2500–3000	2340 —
Bronze-leaded (overplated)	2700					4500–5000**	3000 1500	2560 —
Aluminum alloy	5500	7300–9300	6700–7600	—			1800	—
Silver (overplated)	7500	12000+ 10000–11000	11200–12300	5000			4800– 5000	4260

* Bronze back.
** Bronze back gridded and filled.
1 Cleveland Graphite Bronze.
2 National Advisory Committee for Aeronautics—Technical Note No. 1108.
3 Wright Aeronautical Corporation.
4 Arthur F. Underwood—General Motors Corporation.
5 Albert B. Willi—Federal Mogul Corporation.
6 Bernard J. Esarey—American Brake Shoe Company.
7 American Iron and Steel Engineer, Cleveland Graphite Bronze, Pratt & Whitney Aircraft.
8 Product Engineering Bearing Survey.

metal takes the mechanical loads; the soft lead allows for dirt absorption and also bleeds out if local heating takes place, laying down a thin film of soft metal over the base material.

Gridded Materials

The surface of a hard bearing material is mechanically indented or gridded. A soft metal such as babbitt is cast into the grids. This, like the other methods, provides an escape for dirt and improves bearing characteristics.

These three principal constructions are shown in Fig. 1. Each method overcomes the shortcomings of hard bearing materials. No one method appears to be better than the others.

Thickness

Backs for high-load, high-speed engine bearings have become thinner as loads have increased. In part, this has come about by method of manufacture: Backs made from strip steel to which the bearing liner has been bonded cannot be formed to small radii if the thickness is so great that the metal is ruptured during the forming operation.

Modern bearing backs vary in thickness from 0.062 in. for small diameters to 0.125 in. for 5 in. diameter. Liner thickness also has been reduced and varies from 0.002 to 0.005 in. for micro bearings depending on bearing size and material. On overplated bearings, the thickness of the overplate ranges from 0.001 to 0.005 in. The effect of lining thickness on bearing life is shown in Fig. 2. A small reduction in the liner thickness can increase bearing life many times. As shown in Fig. 2 from 0.014 to 0.032 in., lining thickness has no effect on the bearing life. In the critical zone where thickness varies from 0.005 to 0.014 in., there is a noticeable increase, but the ultimate is reached in the 0.001 to 0.005 in. range.

Another point in favor of thin overlays of soft material on a harder bearing structure is its effect on dry friction. At times when lubrication is not perfect, a low coefficient of friction may enable the bearing to operate until an oil film is restored. In Table I are given comparative values of the dry coefficient of friction, based on an arbitrary index of 1.00 for steel on steel, for several bearing materials rubbing on steel.

In discussing thickness of liners, it should be pointed out that the data given is for high-speed, high-load bearings only. Hydrodynamic or thick-film bearings, covered in another section, have much thicker liners. However these bearings are lightly loaded compared to engine bearings, hence the soft babbitt is capable of holding up without deformation. In general, as the load carrying capacity of the bearing is increased, it is advantageous to reduce liner thickness.

Load Carrying Capacity

Predicting the load carrying capacity of an engine bearing is the principal concern of the bearing designer. It is

Table III—Engine Bearings—Allowable Unit Pressures for Several Classes of Machinery

PASSENGER CARS	
Tin and lead-base—conventional thickness	1200
Micro—tin and lead-base	1400
Copper-lead—35 percent lead	1600
Copper-lead—precision plated	3000
AUXILIARY DRIVES (This includes generators, superchargers and small high speed auxiliary engines.)	
Tin and lead-base—up to 15,000 ft/min	500
Copper-lead—precision plated—up to 2000 ft/min	2500
TRUCKS AND BUSES	
Tin and lead-base micro	1200
Copper-lead—35 percent lead	1500
Leaded bronze—overplated	1800
Copper-lead—precision plated	2500
Aluminum	1800
DIESEL	
Tin and lead-base	1000
Copper-lead—35 percent lead	—
Copper-lead-silver—28 percent lead	1800
Leaded bronze—overplate	1800
Copper-lead—precision plated	2500
Aluminum	1800
AIRCRAFT	
Copper-lead-silver—28 percent lead	3500
Silver and lead-indium	5000

Table IV—Recommended Clearance Ratios for Engine Bearings

Bearing Material	Ratio*—C/D
Babbitts	0.0005
Cadmium	0.0008
Copper-lead	0.00075-0.0010
Aluminum	0.0010
Silver	0.0010

* Larger for bearings smaller than 2½ inch diameter and smaller for bearings larger than 2½ inch diameter.

also the most difficult job. Although the compressive and fatigue strengths of a bearing material may be known these data cannot be used to predict the load carrying capacity of any one bearing. As listed previously, many operating variables may affect load carrying capacity.

Based on past experience, it is possible to select materials and choose bearing dimensions to set up preliminary design—to be proved out by laboratory and service tests.

PV AND ZN/p

The *PV* factor (rubbing factor) for a bearing is still used by some designers to judge its safety, where *P* is the mean pressure, in psi, during one load cycle and *V* is the rubbing speed of the bearing, in fpm. For any one engine or machine operating at a given speed, there is a maximum load beyond which the bearing cannot be operated without breakdown in the oil film or unsafe oil film temperatures in the bearing. This factor is predicated on a number of engine characteristics that include; pattern of the load cycle, lubrication system, and shaft stiffness.

The same bearing used in another engine may or may not be able to take the same load depending on whether the operating conditions are more or less severe. For this reason *PV* values for engine bearings have little significance except when applied to a specific type or line of engine for which sufficient operating data exist on bearing load capacity as affected by changes in machine characteristics.

The ZN/p bearing factor, previously

Table V—Representative Finishes Obtained by Machining or Grinding

(A) Finish	Microinches*	
	h max	h rms
Rough machine surface	10^4 to 10^5	10^3
Medium machine surface (rough grind—furrows visible)	2000 to 10,000	>400
Fine machine (fine ground—furrows invisible)	200 to 400	30 to 100
Honed	40 to 80	2 to 10
Lapped, polished	2 to 40	<4
Fine metallographic finish	<10	<2

* Measured with surface analyzer.

(B) Finish	Microinches		
	Taper Section	Surface Analyzer	Profilometer
Ground	15 to 85	10 to 72	1.6 to 12
Lapped	50 to 90	30 to 48	3 to 6
Sand paper	15 to 30	10 to 24	4.3 to 5
Hydrolap	30	30	2.5
Superfinish	0.5	16	1.6

Table VI—Influence of Journal Finish on Bearing Load Capacity

Type of Finish	rms		Sommerfeld No. at Point of Min. Friction	Relative Load Carrying Capacity
	Transverse	Parallel		
Turned	130	50	0.38	1.00
Ground	70	70	0.25	1.52
Ground	50	50	0.06	6.34
Grit blasted	30–50	—	0.12	3.16
Ground	14	—	0.0017	224.00
Ground	13	—	0.0013	292.00
Chemically etched	13	—	0.0240	15.80
Ground	7	8	0.0012	316.00
Ground	4	6	0.0010	380.00
Superfinished	1	—	0.0008–0.0011	475–345

Fig. 2—Effect of liner thickness on life of engine bearing.

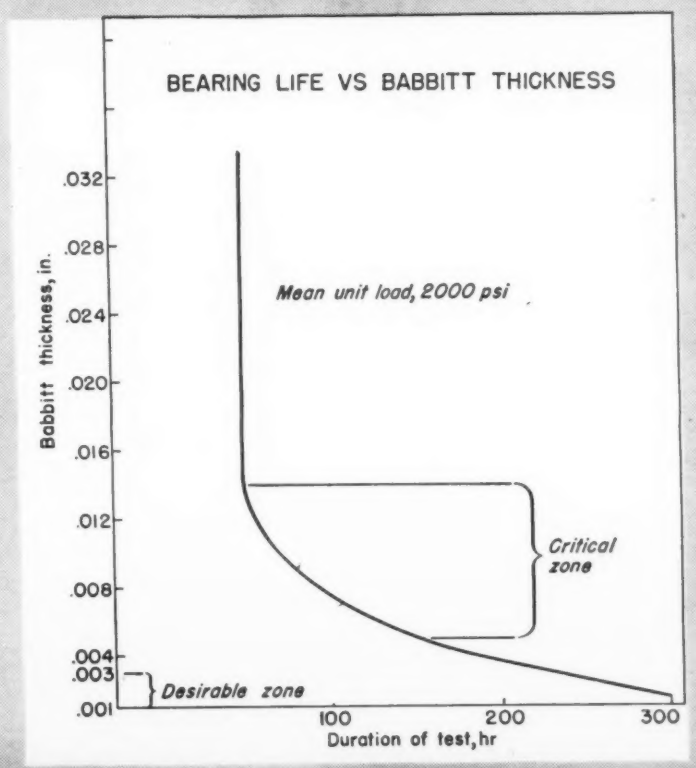
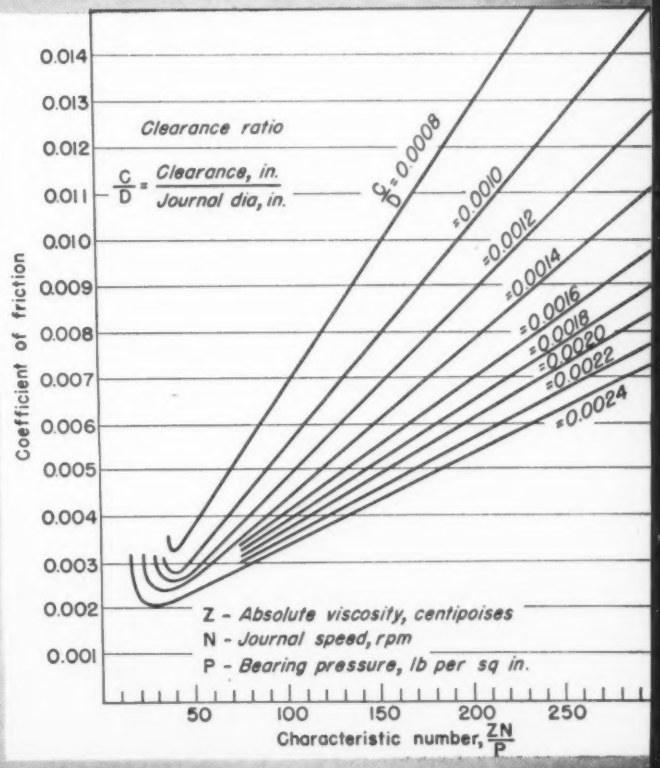


Fig. 3—Effect of bearing clearance on friction f and ZN/p.



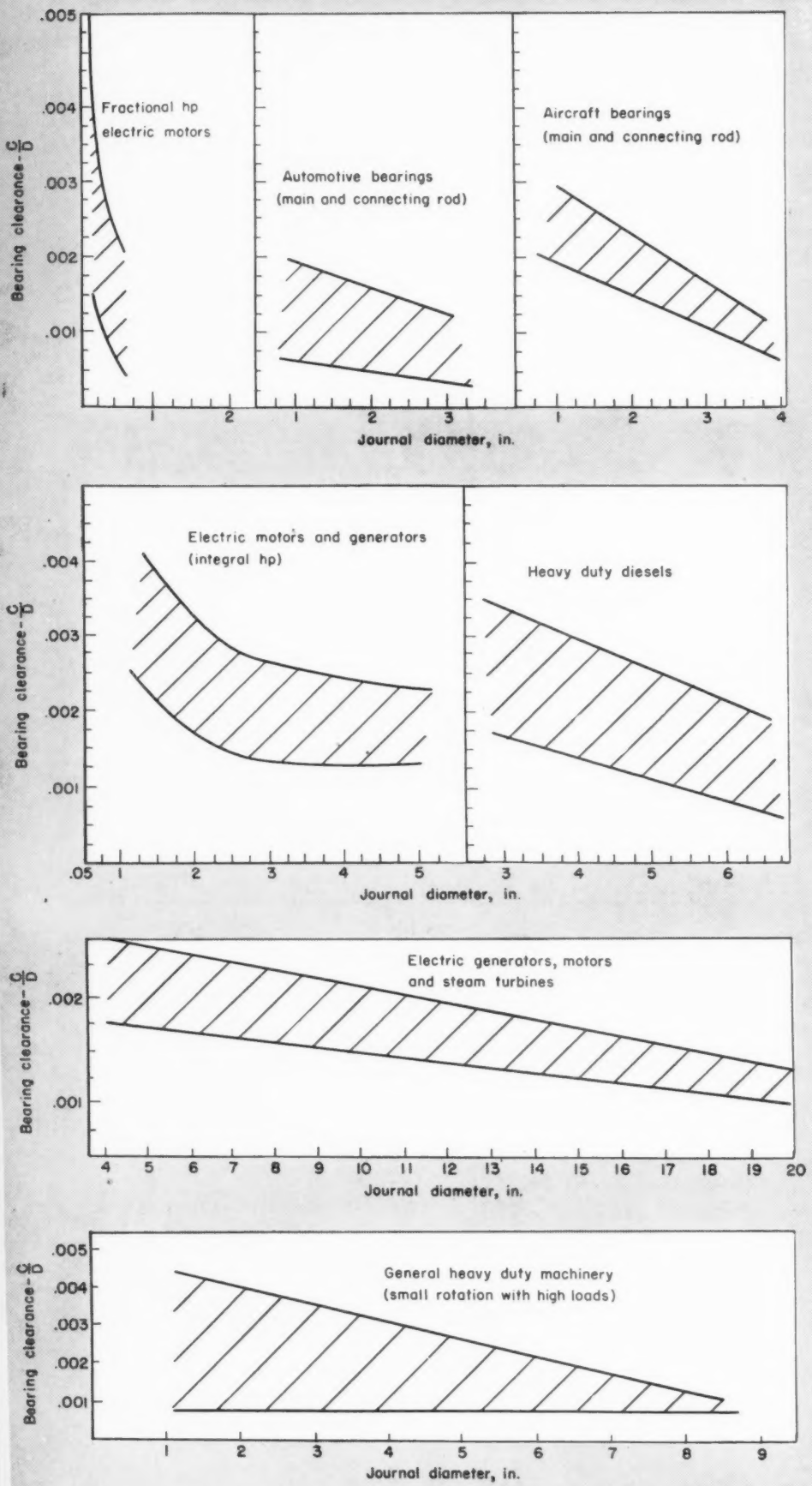


Fig. 4—Current practice for bearing clearance ratios for several classes of machinery.

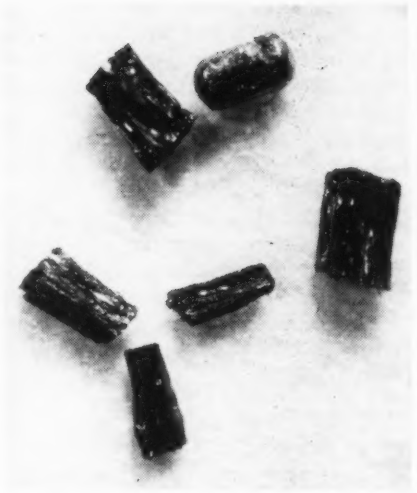


Fig. 5—Hard particles torn from journal and lodged between journal and bearing reduce bearing life.

discussed, is considered a more accurate index of bearing safety. For hydrodynamic bearings, it can be used with some assurance, but for engine bearings the use of a minimum ZN/p may be misleading. However, the use of some criterion is necessary to arrive at any engine bearing design. Some recommended values for PV max and ZN/p min are

Bearing Material	Minimum ZN/p	PV Max	Min S*
Lead-base babbitt	20	35,000	0.050
Tin-base babbitt	10	40,000	0.050
Cadmium base	3.75	90,000+	0.009
Copper-lead	3.75	90,000+	0.009
Silver with lead-indium overlay	2	105,000+	0.005

* Sommerfeld number (see section on hydrodynamic bearings).

Maximum bearing pressures sometimes are used for design purposes for machines in which bearing operating conditions are substantially similar. In Table II, allowable bearing pressures are given for engine bearings. These values have been taken from several sources. The range of values shown indicates that no one value for allowable pressure can be considered as absolute. The figures shown in Columns 1, 2 and 3 are laboratory values and are useful for comparison purposes only. Figures in Columns 4, 5, 6 and 7 are recommended values. Column 8 figures come from the PRODUCT ENGINEERING Bearing Survey. These figures are for connecting rods and main bearings only. Knuckle pin bearings and wrist pin bushings are designed using higher allowables, but they are special types of dynamically loaded bearings.

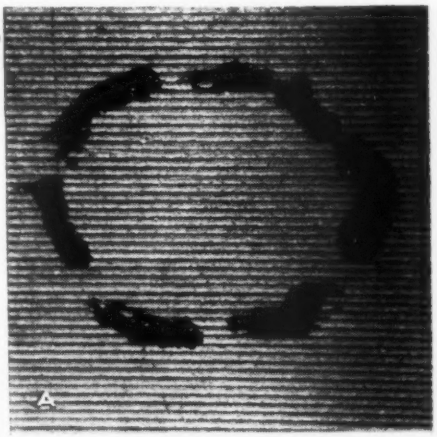
In Table III is given another list of allowable bearing pressures for engine bearings of different materials and several classes of machinery.

Clearance Ratios

Clearance ratio is usually given as C/D where C is the difference in journal and bearing diameters in in., and D is the diameter of the journal. Theoretically there is an optimum value for C/D for every bearing, determined by the conditions of operation of the bearing. Based on theory alone, it is impossible to specify C/D value for engine bearings. In practice relatively large changes in C/D produce little effect on the friction or load carrying characteristics of these bearings. The reason for this can be explained from Fig. 3 where the coefficient of friction is plotted against the characteristic number ZN/p for one bearing. Here C/D has been varied from 0.0008 to 0.0024 in./in.; yet in the transition region from thick to thin-film lubrication, the effect on the ZN/p minimum value is small. For most high-speed, heavy-duty bearings, variation in coefficient of friction is not important.

Also in practice, it is difficult to hold clearances to limits required by hydrodynamic theory. In Table IV are shown clearance ratios recommended by several bearing manufacturers.

These recommended clearances, however, are not adequate to describe



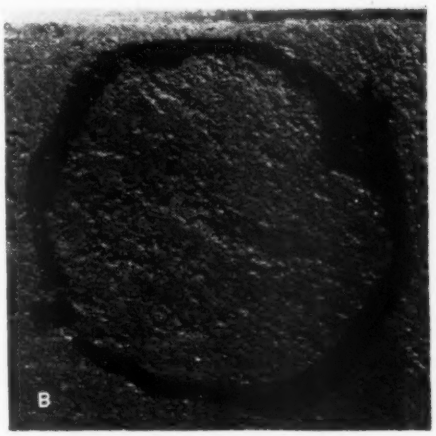


Fig. 6—Magnified view of bearing surface. (A) Machined finish (B) Surface after run-in period.

Table VII—Shaft Conditions Currently Specified for Various Classes of Bearings

Bearing	Where Used	Shaft Material	Finish Operation	Finish	Hardness
Lead base babbitt	Electric motor rotor shaft	ASTM Spec. A235–40T Class E	Ground	——	150 Brinell
	Diesel connecting rod	SAE 1045	Ground	20–25 rms	Rockwell C-25
	Gas engine connecting rod	SAE 1045	Ground	20–25 rms	Rockwell C-25
	Gas engine main bearing	SAE 1045	Lapped	——	275 Brinell
	Gas engine main bearing	SAE 1050	Lapped	——	500 Brinell
	Gas engine main bearing	SAE 1055	Lapped	10–25 rms	500–550 Brinell
	Electric motor rotor shaft	SAE 1040	Ground	——	150–180 Brinell
	Reduction gear-pinion	SAE 1055 SAE 1065	Ground	——	200 Brinell
	Generator main bearing	SAE 1050	Lapped	——	——
	Diesel main bearing	SAE 1055	Lapped	10–25 rms	500–550 Brinell
Cadmium	Transmission planet gear	——	Lapped	10 rms	Rockwell C-55
Copper lead	Diesel main bearing	SAE 1045	Polished	——	Rockwell C-50–55
	Gas engine connecting rod	——	Lapped	10–20 rms	Rockwell C-45
	Diesel connecting rod	SAE 1045	Lapped	15–25 rms	Rockwell C-50
Tin base babbitt	Steam turbine journal	SAE 1045	Turned and polished	——	150–175 Brinell
	Steam turbine journal	Carbon steel nickel steel chrome-moly steel	Turned and polished	——	150–240 Brinell
	Electric generator main bearing	Low carbon steel	Lap	——	Annealed
Bronze	Tractor transmission idling gear	SAE 1010	Ground	——	Rockwell C-60–65
	Aircraft engine rear bearing propeller	AMS 6415	Lapped	8 rms	Rockwell 15M-86
	Aircraft engine connecting rod knuckle pin	AMS 6470	Lapped	8 rms	Rockwell 15N-92
	Aircraft engine supercharger impeller shaft	AMS 6264	Lapped	8 rms	Rockwell 30N–77–80
	Electric motor rotor shaft	SAE 1045	Ground	——	Rockwell C-21
	Electric motor rotor shaft	SAE 1045	Polished	15–25 rms	Rockwell B–75–80
	Gas engine main bearing	SAE 1055	Lapped	10–25 rms	Brinell 500–550
	Aircraft engine planet carrier	SAE 4820	Ground	20 rms	Rockwell C–31–35
Aluminum	Aircraft engine generator drive shaft	AMS 6470	Lapped	8 rms	Rockwell 15N–92
Silver	Aircraft engine master connecting rod	AMS 6415	Lapped	——	Rockwell 15N–86
	Aircraft engine connecting rod	AMS 6415	Polished	——	Vickers 500 (10 kg)
	Aircraft engine main crankshaft	AMS 6260	Lapped	——	Rockwell C–58
	Aircraft engine supercharger drive shaft	AMS 6250	Lapped	——	Rockwell C–60
Cast iron	Press connecting pin	SAE 1045	Ground	——	Annealed
Steel (SAE 1020)	Plate shear connecting pin	SAE 4615	Ground	——	Case hardened

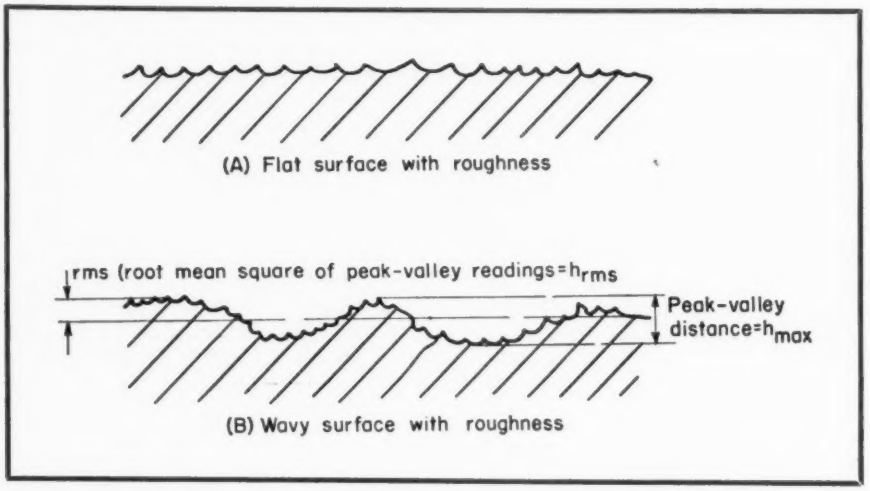

(A) Flat surface with roughness
rms (root mean square of peak-valley readings=h_{rms}
Peak-valley distance=h_{max}
(B) Wavy surface with roughness

Fig. 7—Exaggerated profile of machined surface. Wavy surface may be result of tool support deflections. Bearing and journal surfaces should be smooth for long life.

current bearing design practice for different classes of machinery. In Fig. 4, C/D values for a number of successful bearings used in several classes of machinery are shown plotted against journal diameter. Note that only the automotive, aircraft and diesel bearings are engine bearings. The C/D values for the remaining classes of machinery are shown here for convenience only. C/D curves for automotive aircraft and electric motor and generator bearings are fairly representative. The C/D curves for fractional horsepower electric motors, diesels and heavy electric equipment and turbines are not as reliable. For these, plotted C/D values showed a wide scatter. But the C/D limits shown are more indicative of current practice than the usual blanket recommendations for clearance.

The C/D values shown are admittedly arbitrary, but experience indicates these values have more significance for the engine bearing designer than clearance ratios obtained by using hydrodynamic theory.

Bearing and Journal Condition

Of equal importance with materials used in engine bearings is the surface condition of journal and bearing. To maintain a thin oil film between the bearing and journal on high-speed, high-load engine bearings, all conditions must be favorable. From a practical standpoint it does no good to select bearing and journal material carefully and then neglect the surface of these elements.

When babbitts are used, surface condition is not so important. Heat-treated medium carbon steel journals, ground or ground and lapped are satisfactory. The conformability of the bearing to the journal is good. To a certain extent this is true of cadmium alloy bearings. But when the harder bearing materials such as copper-lead, leaded-bronze, aluminum and silver are used, shaft hardness and surface condition of bearing and journal become critically important.

To prevent scoring and seizing, it is necessary to go to harder and smoother journals and smoother bearings. Journals for high-speed, high-load engine bearings are now finished down to 2-10 rms and are hardened up to Rockwell C-65. Bearing finishes have gone from approximately 40-100 rms down to 2-10 rms.

Table VIII—Current Finish Operations for Engine Bearings

Bearing Material	Finish Operation
Tin-base babbitts	Bored Bored and hand scraped
Lead-base babbitts	Bored and scraped Broached Diamond bored
Cadmium alloys	Machine bored
Copper-lead	Machine or diamond bore Broached
Bronze	Diamond bore Diamond bore and plate Hone, lap, burnish
Bronze (porous)	Machine bore Diamond bore Size, burnish, broach
Aluminum	Diamond bored
Silver	Diamond bore and plate
Steel	Coarse Hone
Cast iron	Coarse Hone

High Spots and Weld Nuggets

The effect of hardened shafts on bearing performance is not clear. The physical nature of present bearing metals is one of hard particles in a softer matrix or soft particles in or on a hard matrix. During operation, at times when the oil film is broken, hard bearing particles (high spots on bearing) rub on the journal. Friction temperatures that result are high enough to cause welding of the bearing to the journal at these local points of contact and the welds are loaded in shear as the journal rotates. These weld nuggets then fail by tearing out at the weld base in the journal, or in the bearing; or by shearing at the weld surface. As a result, hard particles may be torn out of the journal or softer particles out of the bearing. By hardening the journal, the likelihood of tearing the weld nugget out of the journal is lessened, and if the particle is torn out of the bearing it may embed in the soft bearing material before it can do any damage. Fig. 5 shows an

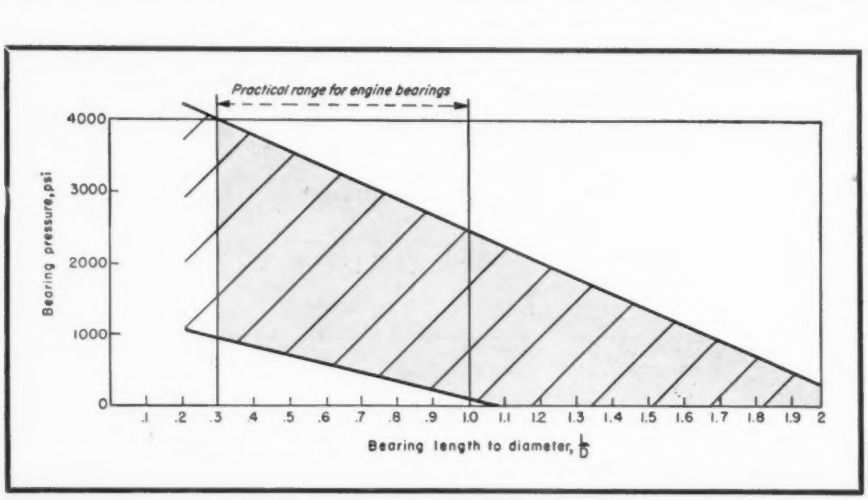

Practical range for engine bearings
Bearing pressure, psi
4000
3000
2000
1000
0
.1 .2 .3 .4 .5 .6 .7 .8 .9 1.0 1.1 1.2 1.3 1.4 1.5 1.6 1.7 1.8 1.9 2
Bearing length to diameter, $\frac{L}{D}$

Fig. 8—Length-to-diameter ratios for current sleeve bearings versus design pressures.

enlarged view of some weld nuggets from a failed bearing. Note that one particle is round as a result of being rolled between bearing and journal.

Fine shaft and bearing finishes reduce metal-to-metal contact, so local welding and resulting scoring is also reduced. It is important, then, to use hard, smooth shafts with the harder bearing materials.

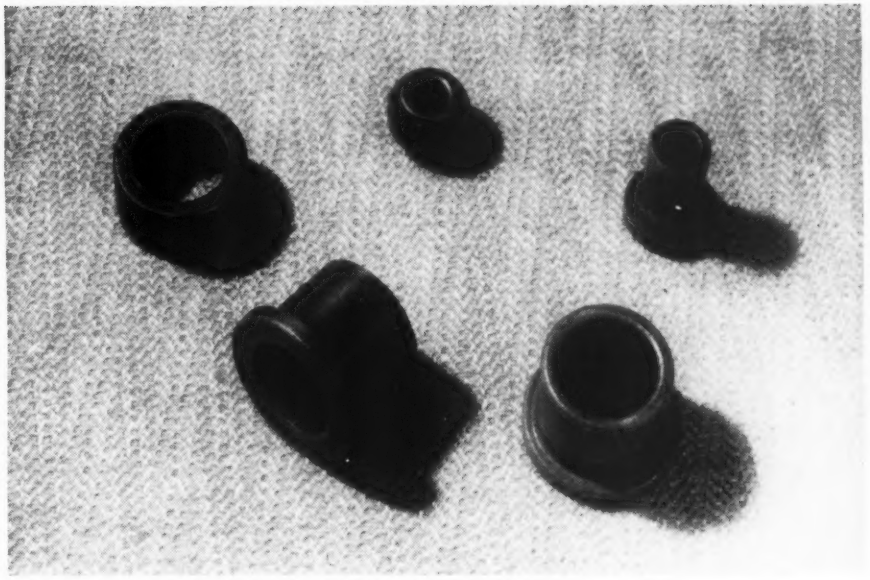
Journal and bearing finishes are obtained by a variety of finish operations, producing surfaces visually smooth but actually quite rough if magnified (refer Fig. 6). Representative machine finishes are given in Table V. Experiments have shown that bearing load capacity is sensitive to surface finish. Table VI illustrates the important effect of finish on bearing performance.

Current design practice for journal material, finish and hardness is shown in Table VII. The data shown were taken from the PRODUCT ENGINEERING Bearings Survey. Finish operations for bearings are shown in Table VIII. Although there is considerable spread in the data shown, it is clear that the trend is to harder journals and smoother bearing and journal surfaces.

LENGTH TO DIAMETER RATIO

L/D ratios (for engine bearings) vary greatly. From the hydrodynamic standpoint, larger L/D ratios are better because they cut down end losses and increase bearing load capacity. On the other hand, it is practically important to cut down bearing length because narrow bearings allow stiffer shafts and smaller machine dimensions. Precision bearings, with a liner a few thousands thick, benefit from stiff shafts and low L/D ratios. In these, a small amount of shaft deflection may be enough to bring about metal-to-metal contact between shaft and bearing. The liner is destroyed and the steel back then rubs on the shaft—usually a damaging condition.

Published values for L/D ratios for various classes of machinery are higher than the L/D values for bearings in current use. The only area of agreement is in the region for thick-film bearings where the bearing materials have little or no effect on bearing dimensions. Results of the PRODUCT ENGINEERING survey are shown plotted in Fig. 8. The range of L/D values for bearings recently designed and in current use are shown. Modern engine bearings running at high speeds and loads are seldom designed with an L/D value greater than one; the significant fact is that the trend in engine bearings is towards smaller L/D ratios, although it does not always follow that highly loaded bearings should have low L/D ratios.



Porous Bearings

AMONG THE VARIOUS TYPES of plain sleeve bearings, porous bearings, made by the powder metallurgy process, have become increasingly popular. Powder mixtures of metals or metal alloys are compressed under high pressures into shape, then sintered in a controlled atmospheric furnace. Sintering temperatures are below the melting point of the alloy or mixture. The resulting bearing structure is porous but strong. By impregnating with a nongumming lubricant, a bearing is made that requires no external source for lubrication, provided bearing loads and speeds and other design conditions are not too severe.

Porous bearings are low in cost, quiet and are used for applications where it is inconvenient or impossible to provide or maintain lubrication by other means.

The theory of lubrication for these bearings is not clear. During normal operation, there appears to be a molecular layer of lubricant on the liner surface that is capable of supporting moderate loads at medium speeds.

Many metals are used for porous bearings. The two principal porous bearing types have a bronze base or an iron base. In Table I, SAE specifications for these two types are given.

In addition to these basic types, there are aluminum, nickel and brass porous bearings. In all there are a large number of proprietary porous bearing materials. Several manufacturers prefer to

Table I—Composition of Porous Metal Bearings

Elements	Type I		Type II	
	Class A Copper-Tin	Class B Copper-Tin-Lead	Class A Iron-Carbon	Class B Iron-Copper
Copper	87.5–90.5	82.6–88.5	—	5.0–30.0
Iron, min	—	—	95.0	Remainder(2)
Tin	9.0	9.0	—	
Lead	—	2.0–4.0	—	
Zinc, max	—	0.75	—	
Nickel, max	—	0.35	—	
Antimony, max	—	0.25	0.5–2.0	
Carbon	1.5 max(1)	1.5 max(1)	3.0	
Other, max	0.5	0.5	—	3.0

(1) Graphite.
(2) Total iron and copper not less than 97 percent.

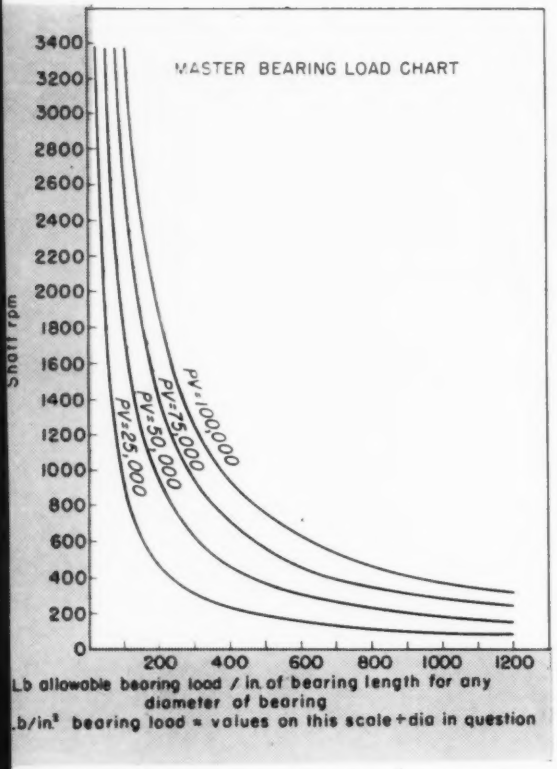


Fig. 1—PV factors for porous bearings.

use no graphite in their bearings, others make porous bearings from a mixture such as brass and graphite. This latter type is a true "oilless" bearing material with the graphite acting as a lubricant.

Like the heavy-duty, high-speed engine bearings, there appears to be no way to predict accurately porous bearing performance. Load, speed, location, clearance and lubricant influence the load carrying capacity and life of any one bearing in any one material. However, manufacturers of porous bearings have checked the performance of their bearings for many uses. From this they have set up conservative performance criteria in terms of temperature, load and pressure for various types and designs of porous bearings.

Oil film temperature is one of the most important control factors determining bearing performance. If temperature is too high, the lubricant may bleed out and leak from the bearing. Also if lubricant viscosity drops too much, a film cannot be maintained. For porous bearings not lubricated from outside sources, all the friction heat must be conducted away through the heavy metal and bearing housing, so the equilibrium temperature for any set of operating conditions must be limited in order to assure bearing life.

The impregnating temperature for most porous bearings ranges from 170 to 220 F and safe maximum operating temperature is about 150 F. If the housing in which the porous bearing is pressed, is heated from some other source than the bearing, the performance of the bearing is adversely affected.

In general, porous bearings are not run continuously at speeds below 75 or over 1,500 fpm. At low speeds, the oil film may be broken and at high speeds, high oil temperatures will produce the same result. Assuming fairly stiff shafts, safe temperatures and no unusual load or speed conditions, safe load and speed combinations (*PV* factors) can be set for any one bearing material. Allowable shaft speeds and bearing loads for bearing materials with *PV* factors from 25,000 to 100,000 are shown in Fig. 1.

Representative *PV* values for various porous bearing materials are

Bronze (general purpose)	50,000
Bronze (severe pressure)	35,000
Iron (normal duty)	35,000
Iron (light duty)	75,000

Outside Lubrication

If porous bearings are lubricated by wick feeding or pressure lubrication, their performance is better to a degree depending on the nature of lubrication. Oil flowing through the bearing carries away some of the heat that would otherwise be conducted through the bearing material. For any bearing installation then, equilibrium temperature will be lower and bearing load capacity and life increased greatly.

The effect of lubrication on load capacity of a porous iron bearing with a specific gravity of 6.4 run at medium speeds is shown in Fig. 2. Tests were started with full lubrication with oil supplied at 68 F and 15 psi inlet pressure. Oil clearance was 0.010 in. After a run-in period, the type of lubrication was changed. Oil film temperatures did not exceed 140 F for all tests.

Test results indicate that porous bearings can carry loads equal to solid bearings if sufficiently lubricated, and provided such loads are steady and unidirectional.

The ductility of most porous metals is low. They cannot be used for heavy-duty, high-speed applications. Typical applications are for automobile pedal bushings, water pumps, generator and distributor shafts.

Clearances and Tolerances

Porous bearings are made with good surface finish and bore tolerances of ± 0.002 in. If closer tolerances are required the bearings are sized. Sizing narrows the bore tolerance to ±0.0005 in. and improves surface finish.

As a rule, porous bearings are not machined because machining tends to close the pores of the metal. If press fitted, the final ID of the bearing is smaller than the initial ID and the difference must be taken into account in the selection of a bearing or the design of the shaft running in the bearing. Recommended clearance between bearing and shaft is 0.0007 to 0.0012 in. per in. Smaller clearances are likely to lower bearing life.

In Fig. 3 nominal clearances are shown for precision equipment, electric motors and machine tools. Nominal press fit allowance is shown also.

Fig. 2—Effect of lubrication on capacity of porous iron bearing.

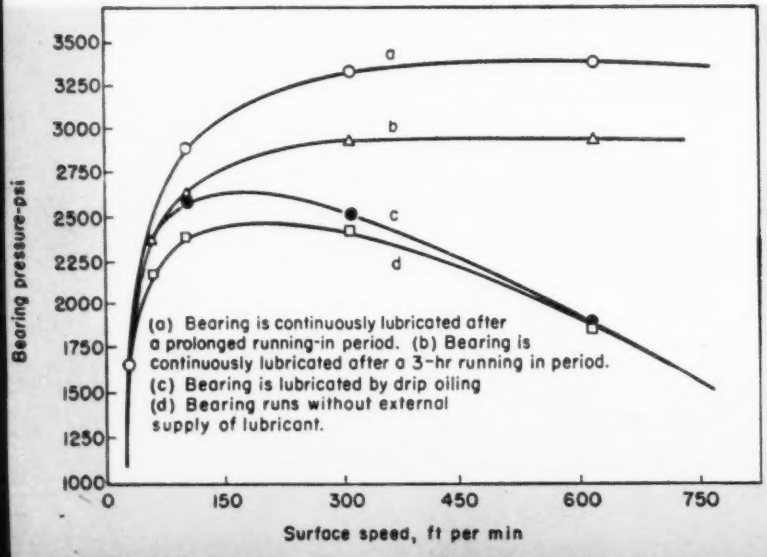
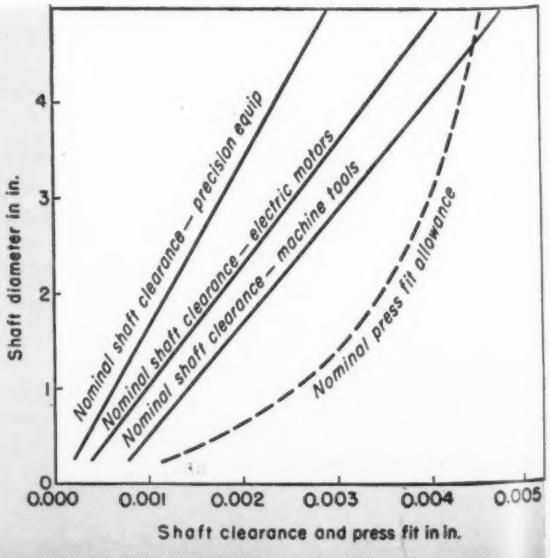
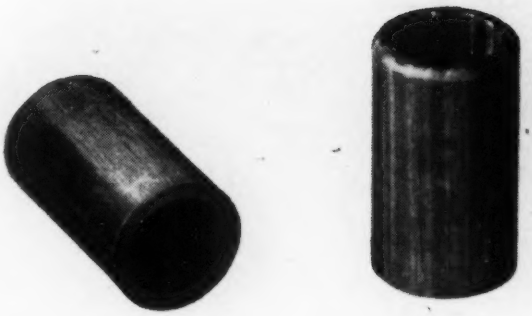
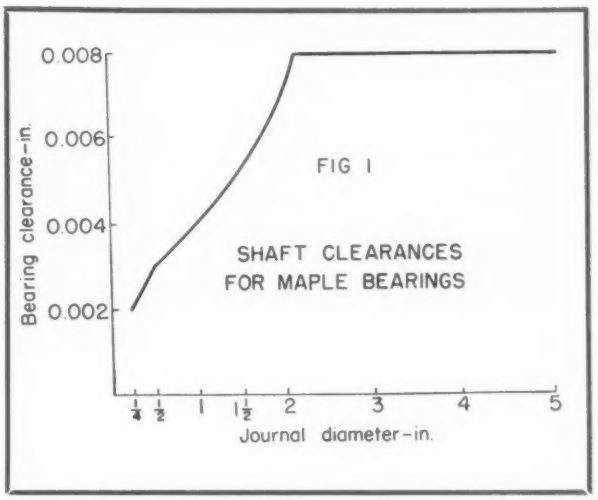


Fig. 3—Shaft clearances for porous bearings.







Wood Bearings

A LARGE NUMBER of wood bearings are used in industrial equipment. Of these, most are made from lignum vitae, rock maple or oak. The chief advantages of wood bearings are low cost and cleanliness. Wood bearings never score or seize the journal.

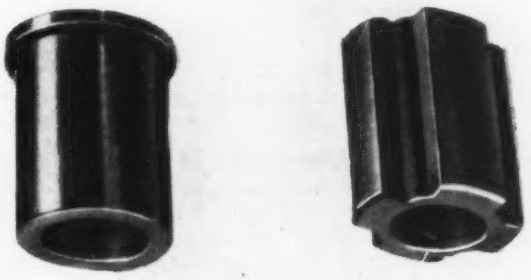
Wood bearings, like porous metal bearings, can be used when there is no outside source of lubrication. Lignum vitae contains its own lubricant in the guaiac gum making up one-third of its weight. Other woods are impregnated with oil similar to porous metal bearings. Wood bearings expand or contract with change in moisture content. This movement is not equal in all directions because of the anisotropic nature of wood. So wood bearings are used only where large clearances and large tolerances are permitted. But the ability of wood bearings to perform under adverse conditions with little maintenance is recognized.

Lignum vitae has a specific gravity of 1.35 and machines like free cutting brass. In the past, lignum vitae has been used as a liner for steel mill roll neck bearings. With adequate lubrication it has been used at bearing pressures up to 2,000 psi. The newer laminated phenolic bearings have replaced lignum vitae bearings because of their greater dimensional stability and low coefficient of friction. Normally, lignum vitae bearings are designed for working pressures from 75 to 150 psi in the direction of the long grain and 300 psi if end grain loaded. No data exist to define completely the performance of this material.

The principal limitation on lignum vitae bearings is operating temperature. For long life, equilibrium operating temperatures should not exceed 120 F.

The guaiac gum in lignum vitae is an anti-oxidant and retards rancidity in animal fats. For this reason lignum vitae bearings are used in food processing machinery and dairy machinery.

HARD MAPLE BEARINGS are similar to lignum vitae bearings. The tendency to char coupled with low thermal conductivity limits use to installations where equilibrium temperature is 150 F or less for uninclosed bearings and 220 F or less for inclosed bearings. Fig. 1 gives recommended clearance values.



Carbon-Graphite Bearings

MACHINED AND MOLDED carbon-graphite is used in machinery and equipment where temperature, materials handled, or location in assembly prohibit use of lubricants or prevent proper maintenance. These bearings are: (1) Self lubricating, (2) chemically inert, (3) able to withstand high temperatures and (4) dimensionally stable over a wide temperature range.

These four characteristics result in their use in conveyor rolls, chemical sprays and dips, immersion rolls in the textile and paper industries, liquid meters, electric clocks, pumps handling corrosive liquids, beverages, foods, and water, and in several automobiles in the clutch throw-out assemblies.

The addition of a metal or metal alloy to the carbon-graphite composition greatly improves its physical properties, particularly transverse and compressive strengths and density. Design data for carbon-graphite bearings are given in Table I. (See next page.)

In designing for this type of bearing, no provision for bearing cooling is necessary. As long as the ambient temperature is below the critical temperature of bearing material used, no bad effects will result. Carbon-graphite bearings require hard smooth journals. Rough journals abrade the bearings, causing excessive wear.

Table I—Design Data for Carbon-Graphite Plain Sleeve Bearings

	Carbon-Graphite Grades (Non-metallically Treated)	Carbon-Graphite Grades (Metallically Treated)
Composition	Carbon-graphite	Carbon-graphite treated with either babbitt, copper, cadmium, lead-bronze or silver
Max. Pressure	20 psi wet or dry	350 psi dry, 600 psi wet
Clearance	Inside diameter, in. / Normal running clearance,* in. 0.187–0.500 / 0.001 0.501–1.000 / 0.002 1.001–1.250 / 0.003 1.251–1.500 / 0.004 1.501–2.000 / 0.005	Same
Temperature limitation	900 deg F in oxidizing atmospheres 5400 deg F in non-oxidizing atmospheres	Dependent upon melting point of metal and atmospheric operating conditions Copper 1981 F Babbitt 464 F Cadmium 600 F Lead-bronze 619 F Silver 1761 F
Speeds	Dependent upon too many variables; high loads require low rpm—low loads permit high rpm, i.e., 50,000 rpm would require loading to be negligible	
Shafting	Cast iron, hard chrome plate, steel preferably 400 Brinell or over, phosphor bronze over 135 Brinell	
Coeff of friction	From less than .10 to .25 dependent on many variables such as rpm, load, wet or dry running, dust, grit, shaft finish, etc.	
Maximum diameter	14 in. OD	14 in. OD
Maximum length	No limitation on length to diameter ratio	

* Applies for steel shaft and operating temperature of less than 200 deg F

Cast Iron and Steel Bearings

THESE MATERIALS are used where conditions of operation warrant or allow. Little factual design data exist to help the designer. In most new applications, past performance is used as a design criterion.

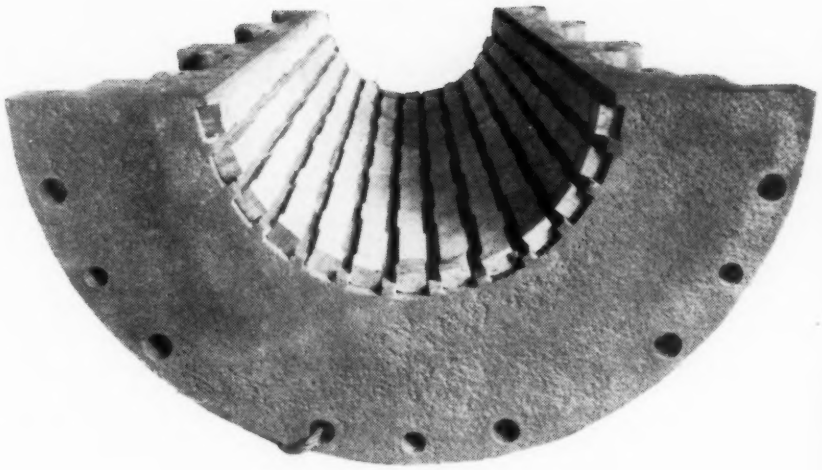
Cast iron bearings are not run at speeds greater than 130 fpm or unit pressures greater than 500 psi. Lubrication must be better and bearing clearances larger than for most other bearing materials. This is done to avoid scoring if hard particles from the cast iron should be torn loose and ride between the bearing and journal. The general rule is 0.0015 to 0.0020 in. clearance per in. diameter of the journal. Journals can be made of soft steel with a Brinell hardness from 150-250, preferably nearer 250. Smoothness of journal and bearing are important bearing life factors. Journals are usually ground and polished; bearings are diamond bored. The maximum operating temperature is about 150 F.

Steel bearings gall quickly with poor lubrication. Galling can be controlled to some extent by using dissimilar steels for the bearing and journal. Nickel steels are used for steel bearings. SAE 2312-2315 steels have been used with some success.

Steel bearings require very smooth surfaces to reduce galling tendencies. For most applications they are designed to operate at low loads and at speeds not over 150 fpm. Because of their poor performance when inadequately lubricated, the operating temperature should not be more than 125 F. Steel bearings should be used to support shafts operating with intermittent motion rather than continuous motion.

Bearings Used in the Electrical Industry

Where Used	Operating Requirements for Bearings	Material
Instruments	Long life, low friction, low wear, resistance to shock	Jewels such as diamond, sapphire, etc.
Timing motors	Low speed, high loads, low wear, long life	Copper, bone, fiber or bronze
Switch gear	Very high pressure, negligible lubrication	Bronze, cast iron, thermosetting plastics
Fractional hp motors	Low cost	Tin-base babbitt porous metal, graphite bronze, copper-lead, cast iron, aluminum alloy
Integral hp motors	Long life	Tin-base babbitt, bronze, copper-lead
Turbines, generators, gears	Long life, large factor of safety, low wear at low speeds, resistance to scoring	Babbitts, high lead bronzes (for auxiliary drives)



Plastic Laminate Bearings

ADVANTAGES of plastic laminate bearings are low friction, shock resistance, conformability, embeddability, nonscoring characteristics and strength. They usually are water lubricated but for some installations, the lubricant is grease, oil, or an emulsion of water and grease.

Laminated plastic bearings may be tailored to fit operating conditions. Production variables are

(a) Resin used
(b) Resin content
(c) Filler used
(d) Molding pressure
(e) Molding technique
(f) Construction

Beyond this are other considerations common to all sleeve bearings such as conditions of operation, lubricant, and bearing dimensions.

Almost all plastic bearings are made using a phenolic resin and a cotton duck filler (ranging from 8 oz to 50 oz per sq yd). Some laminates have a metallic oxide as an additional filler to improve flat-wise wear characteristics of laminates having fine weave fillers.

Bearing strength is determined by direction of loading relative to direction of laminations. Some common bearing types are shown in Fig. 1. When load is applied perpendicular to the laminations, the bearing can take high loads. Because of this, the molded segment bearing and the tubular bearing are high load bearings, the rest low load.

Phenolic bearings cannot be run at temperatures much above 200 F. Above this temperature the material begins to decompose thermally.

Recommended load-speed values for the bearing types shown are given in Table I together with the nominal size for each type of bearing.

Laminated bearings are frequently used in installations where the lubricant may contain acids or alkalis. These do not affect the material unless they are highly concentrated.

JOURNAL CLEARANCE

Laminated bearings expand in water and other lubricants. Expansion is not equal in all directions, being greatest perpendicular to the direction of laminations. Also this expansion depends

Table I—Design Data for Molded Plastic Laminate Bearings

Bearing Type	Nominal Pressure, psi	Nominal Speed, fpm	Nominal Diameter, in.	Class of Service
Molded segment	2,500	1,000	18	Mill roll necks, high loads, severe service
Stave	100	400	15	Marine propeller shafts, steady loads
Tubular	1,000	1,000	4	High impact load
Edgewise (A)	10	300	4	Low load and speeds, installations requiring close tolerances
Edgewise (B)	10	300	2	General purpose stern tube bearings for small ships

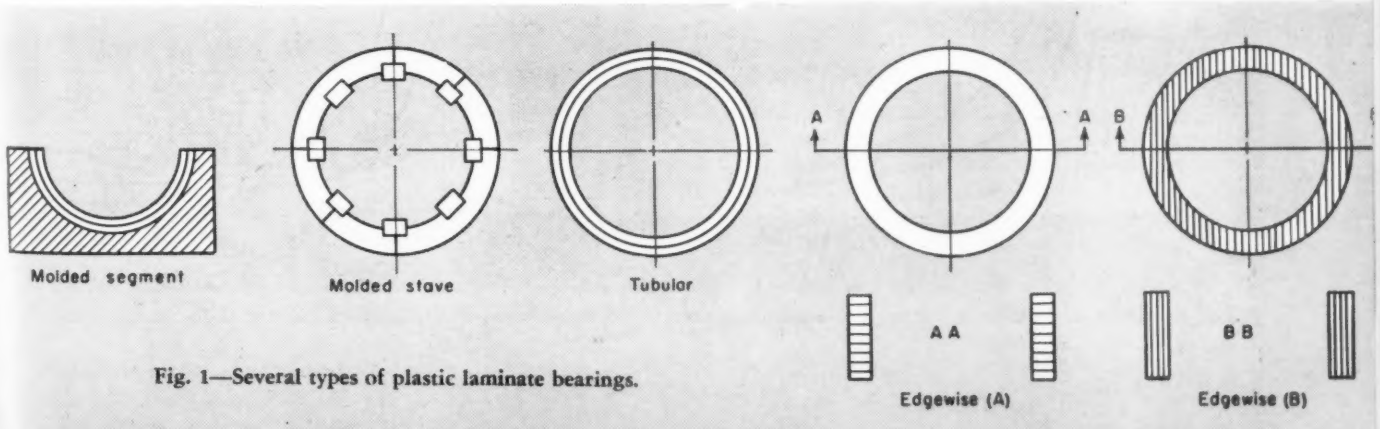


Fig. 1—Several types of plastic laminate bearings.

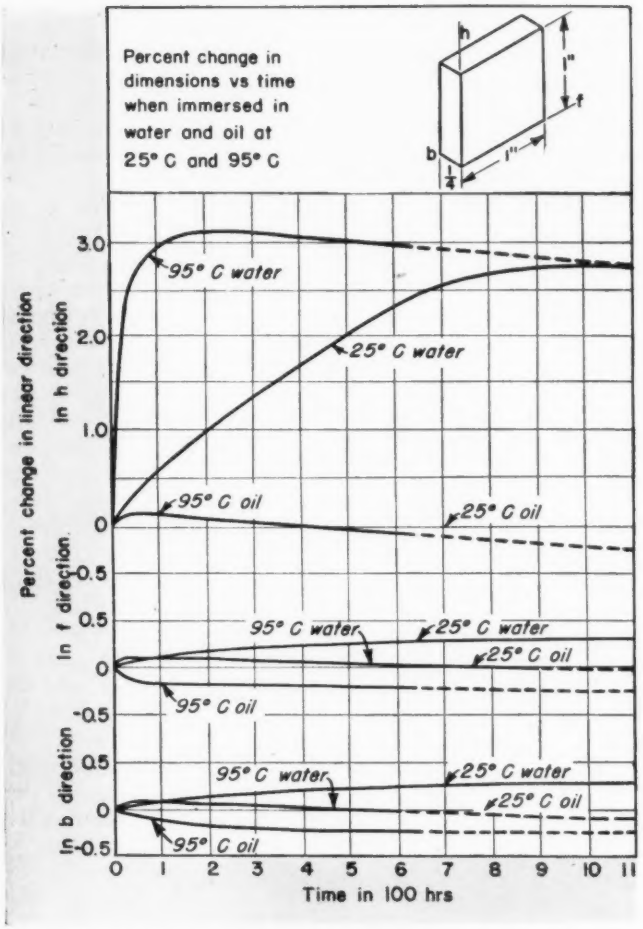


Fig. 2—Expansion curves for one plastic laminate immersed in oil and in water.

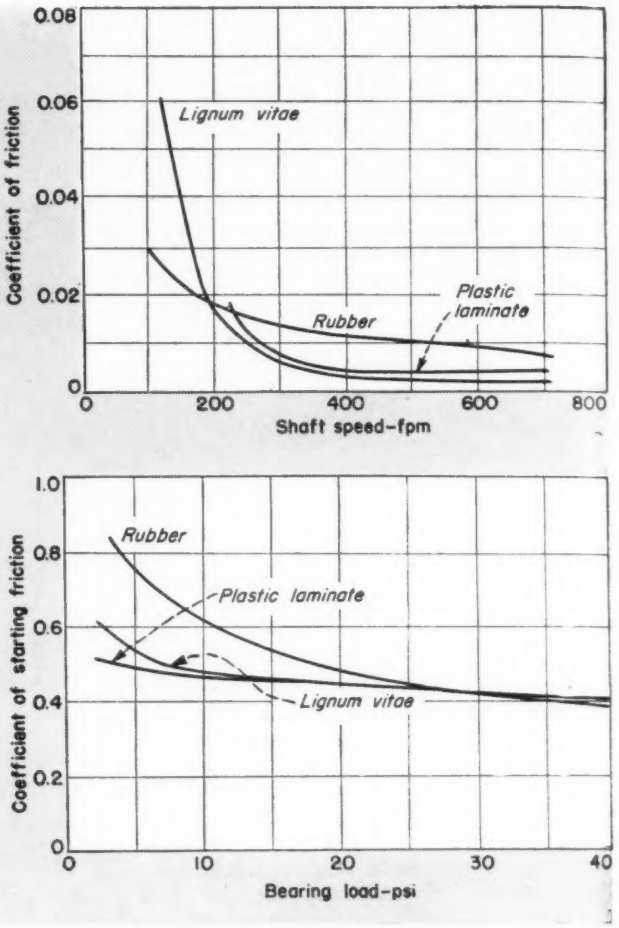


Fig. 3—Coefficients of friction for marine phenolic laminate compared with other materials.

on the lubricant and the time the bearing contacts the lubricant. Typical expansion curves are shown in Fig. 2 for a phenolic laminate with an 8 oz filler and 60 percent resin content, immersed in oil and in water.

Because of varying expansion rate, clearances for phenolic laminate bearings cannot be set until the conditions of service are known. In general, clearances of 0.001 in. per in. of journal diameter are considered satisfactory after swelling takes place.

Friction

Phenolic laminate bearings are used for applications where rubber and lignum vitae are also used. The curves in Fig. 3 compare the friction characteristics of the three materials, and in Fig. 4, the effects of change in bearing pressure or in water temperature on phenolic molded bearing performance are shown.

Journal Condition

Journals should be hard and polished to reduce wear. Cast iron and steel journals wear better than nonferrous metal journals.

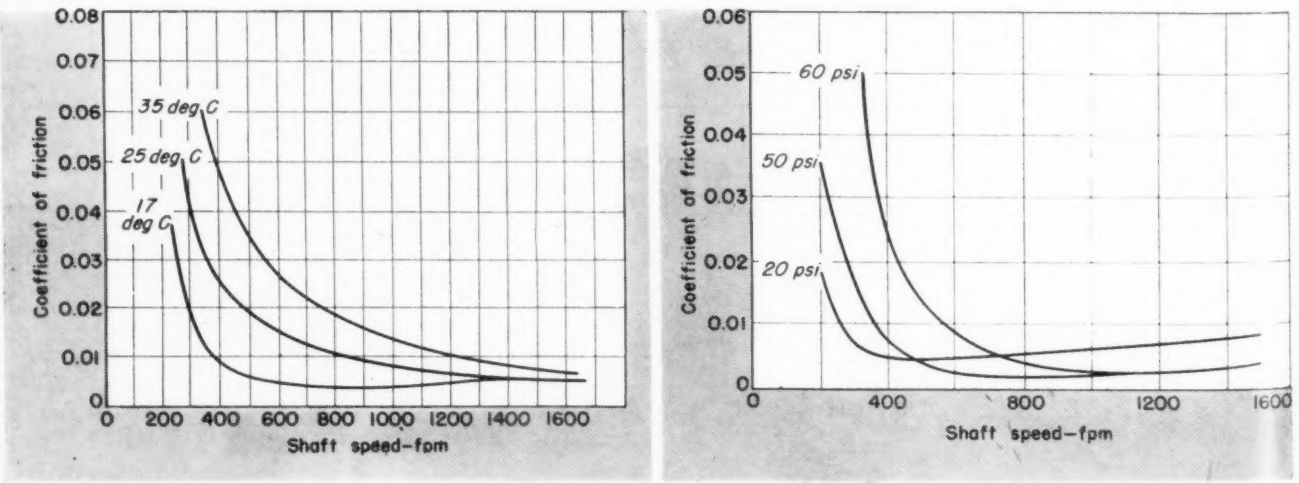


Fig. 4—Effect of bearing pressure and water temperature on coefficient of friction for a marine plastic laminate bearing.

Rubber Bearings

RUBBER BEARINGS are a special class. These are designed specifically for installations where water or some other low viscosity fluid is the lubricant and sand or other hard particles can get into the bearing. Principal uses are for marine propeller shafts, rudder pintles, pumps and hydraulic turbines.

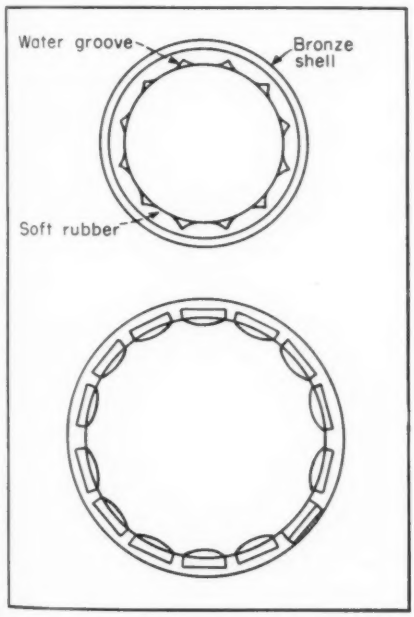


Fig. 1—Two types of rubber bearings.

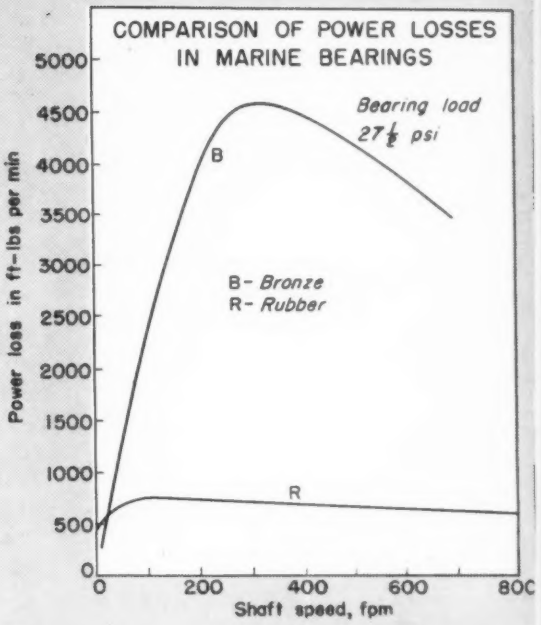
Two types of rubber bearings are shown in Fig. 1. Grooves in the bearing surface allow sand and foreign matter to be carried out of the bearing by the lubricating and cooling water which flows through them.

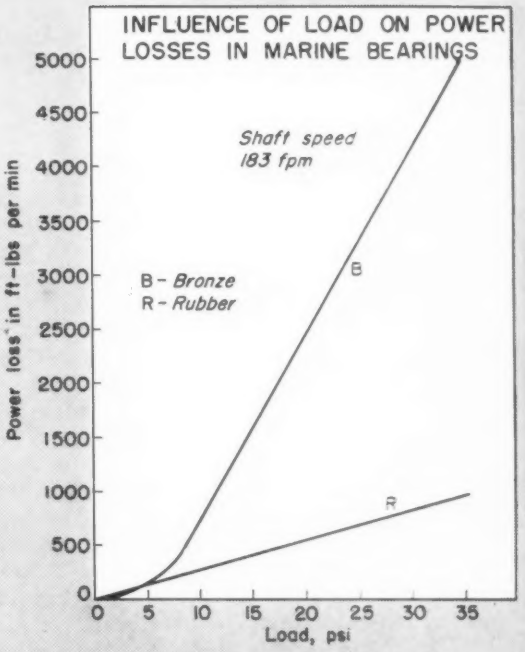
Properly designed rubber bearings have longer life under the conditions for which they are recommended than hard surface bearings, absorb shock and vibration loads and operate quietly.

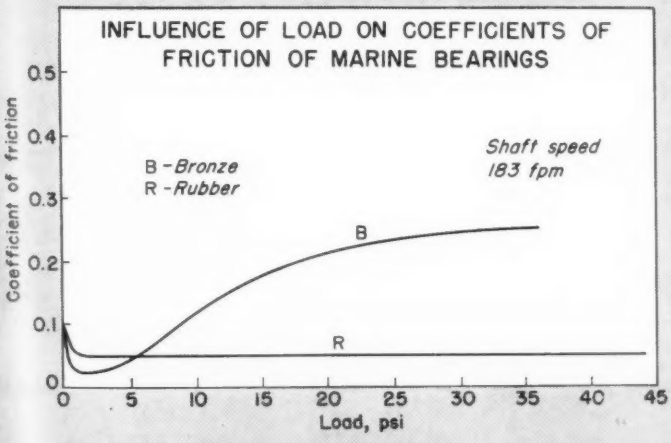
Compared to oil lubricated metal bearings they can support only low loads. For marine use, operating pressures range up to 50 psi. There appears to be no practical limit to journal speeds. Fig. 2 gives the friction and power loss characteristics of rubber bearings compared to bronze bearings.

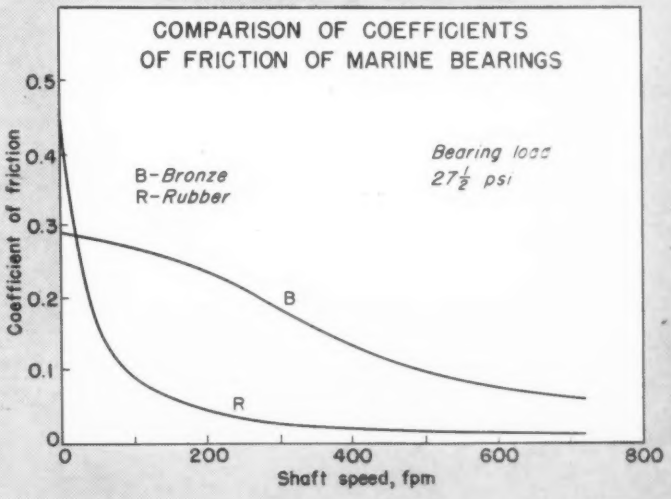
As indicated by the curves, rubber bearings, water lubricated, operate with less friction than bronze bearings and consequently reduce power losses. Since many of these bearings are large, the saving in power through reduced friction is appreciable.

Fig. 2—Performance characteristics of rubber bearings compared to bronze bearings, marine use.









Hydrodynamic Bearings

HYDRODYNAMIC BEARINGS operate in the thick-film region. The load on these bearings is transmitted to them through the oil film separating the bearing and the journal. The main problem in design is to maintain an adequate oil film between the bearing and journal even under adverse operating conditions.

The literature on hydrodynamic bearings is extensive and has as its genesis the work of Reynolds in England and Petroff in Russia who established the concept of hydrodynamic lubrication approximately 70 years ago. Since then the theory has been refined and adapted for the design of many types of hydrodynamic bearings.

But modern hydrodynamic bearing procedure can only be used to design fully lubricated bearings loaded in one direction with a constant load (Recent work by several investigators indicates that this limitation may be lifted in the near future). For this reason hydrodynamic theory is of little use in designing engine bearings or others when loads vary in magnitude and direction. Bearings for turbines, generators and mills, however, are designed using hydrodynamic theory. These bearings, unlike engine bearings, are designed for low power loss rather than high load capacity.

The successive positions of a full journal as the shaft speeds up are shown in Fig. 1, page 131. During normal operations, if a minimum oil thickness, h_o, is maintained, the bearing will run for an indefinitely long period. Theoretically this oil film can be as little as 0.000025 in. thick, but practically it varies from 0.0001 to 0.00075 in. There is no way to select the minimum allowable oil film thickness, h_o, for a new bearing except by comparison with values obtained in other bearings used similarly.

The minimum film thickness then is one criterion for determining safe hydrodynamic operation. The other is maximum operating temperature. If the bearing runs hot, the lubricant will deteriorate, or the bearing material may soften or corrode. The maximum temperature is around 200 F for most hydrodynamic bearings.

MINIMUM THICKNESS

Minimum film thickness is related to operating conditions by the following equations:

$$h_o = \frac{C}{2}(1 - \eta)$$

$$12\pi^2 S = (2 + \eta^2)(1 - \eta^2)^{1/2}$$

where:

h_o = minimum film thickness, in.
C = diametral clearance, in.
S = Sommerfeld variable = $(D/C)^2 \frac{\mu N}{P}$
D = diameter of shaft, in.
μ = absolute viscosity of lubricant, lb sec/in.^{2}
N = speed of journal, rpm
P = projected bearing load, psi = W/LD
W = bearing load, lb
L = bearing length, in.

η = eccentricity ratio (or attitude) = $\frac{\text{journal eccentricity}}{\text{radial clearance}} = \frac{C/2 - h_o}{C/2}$

These equations are shown plotted in Fig. 2. As shown, the film thickness is affected by bearing clearance, diameter, oil viscosity, shaft speed, and load on the bearing. For any one bearing $\mu N/P$ must not be less than the value corresponding to minimum available film thickness.

The foregoing equation takes bearing clearance into account but assumes bearing length to be infinite. For other than very long bearings the value of S taken from Fig. 2 must be multiplied by a correction factor to account for bearing length. This correction factor is shown in Fig. 3. The practical significance of the factor is that narrow bearings cannot be run at $\mu N/P$ values as low as those considered safe for wider bearings. The correction factor curve shown, is conservative for narrow bearings with a high rate of oil flow.

FRICTION

The coefficient of friction for a hydrodynamic bearing also can be expressed as a function of the Sommerfeld number. The relation is shown in Fig. 4. Here f equals F/W where F is the total friction load. Regardless of the individual values of μ, N, or P, D or C, the curve indicates there is only one value for the coefficient of friction for any one Sommerfeld number.

This curve is plotted for an infinitely long bearing. The correction factor for a bearing of finite length is shown in Fig. 3. As shown, the friction for a finite length bearing is somewhat less than for an infinitely long bearing.

OIL FLOW

Oil flow through a bearing is necessary to help maintain the oil film and sometimes cool the bearing. Oil flow computations for newly designed bearings do not always give reliable results but do give the designer some sort of check on his design. There are a number of formulas (see Table I) available to estimate oil flow. Each has limitations. These formulas are basically the same and have been derived for center groove bearings or end-lubricated bearings with unidirectional loading.

In oil flow calculations, if the

amount of oil capable of flowing through the bearing exceeds the amount needed to limit temperature rise, the bearing design is satisfactory. When the oil flow is not sufficient to hold the upper temperature limit, bearing clearance must be increased, or additional grooving added to give the necessary increase in flow.

Load Carrying Capacity Formulas

The load carrying capacity of a hydrodynamic bearing can be gotten for a wide range of operation provided sufficient data are available to compute the heat dissipation ability of the bearing. As previously mentioned, two criteria are used to determine safe hydrodynamic operation.

1. The minimum film thickness, h_o, should never be less than a certain value, this value establishes the minimum value for the Sommerfeld variable, S.

2. The bearing temperature should never exceed a certain value, determined by experience to be safe for the bearing material or for the lubricant used.

A typical safe-load speed curve for a given bearing is shown in Fig. 5. In the region bounded by the curve marked S_{min}, the work done on the bearing in overcoming fluid friction results in an increase in bearing operating temperature, T. The heat created is conducted away by the bearing material and by the bearing lubricant at a sufficiently high rate so that the allowable temperature is not exceeded and the maximum load is determined by the film thickness obtaining. The operating temperature reaches equilibrium when work done on the bearing is equal to heat removed from the bearing. This is expressed as:

$$fPDL\pi DN = B\pi DL(T - T_R)^a + \rho QC(T_2 - T_1)$$

where B and a = empirical constants depending on the ability of the bearing to conduct heat

T = bearing oil film temperature

T_R = ambient or room temperature

T_2 and T_1 = exit and inlet temperatures of lubrication oil

ρ, C, Q = density, specific gravity and flow of oil

In the S_{min} region, the equilibrium temperature is below the maximum allowable temperature T_{max} and bearing load at any speed is only limited by the value for S_{min} selected.

In the region limited by the curve T_{max}, the load at any speed is limited by the maximum bearing temperature selected for safe operation.

To set up the safe values for any one bearing, three other relations are needed. In addition to the heat balance formula shown, these are:

$$P = \left(\frac{D}{C}\right)^2 \frac{\mu N}{S}$$

$f = \Psi(S)$ (refer Fig. 4)

$\mu = \mu_o\beta(T)$ where μ_o is the oil viscosity at a standard temperature and $\mu(T)$ is a function of bearing oil-film temperature.

For any bearing, a load speed curve similar to the one shown in Fig. 5 may be set up. If a safe value for S_{min} is chosen, μ, f and T may be eliminated from the four equations and a

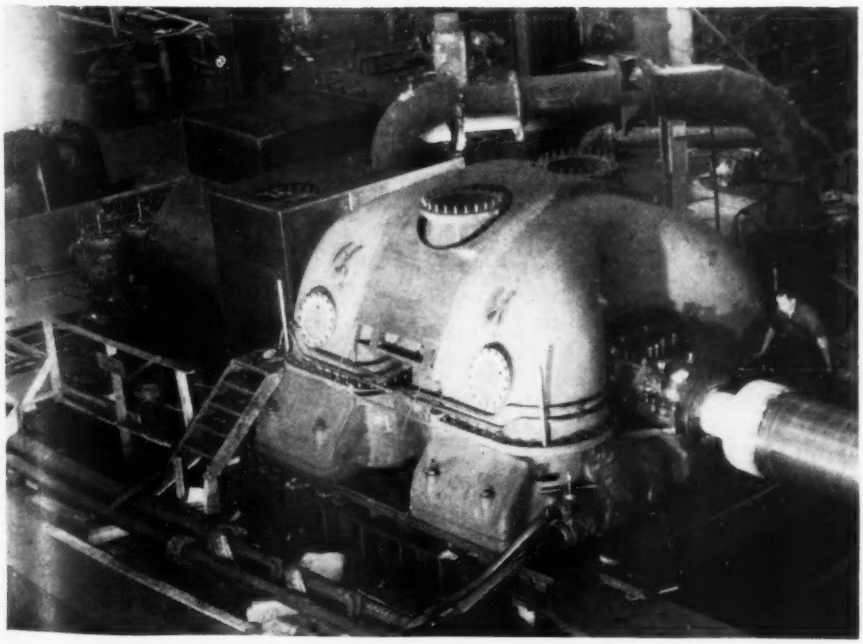


Fig. 1—Large steam turbine equipped with hydrodynamic journal bearings. These bearings are lightly loaded and are designed principally for low power loss.

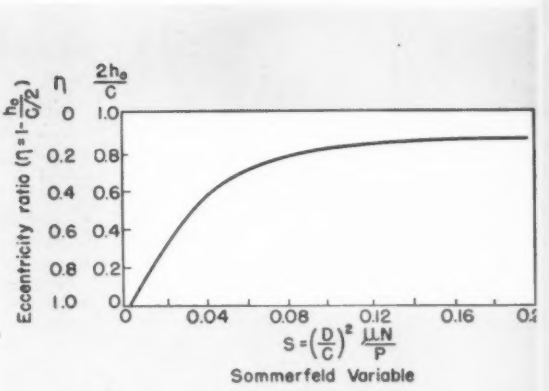


Fig. 2—Sommerfeld variable as a function of bearing eccentricity ratio, η, and film thickness h_o.

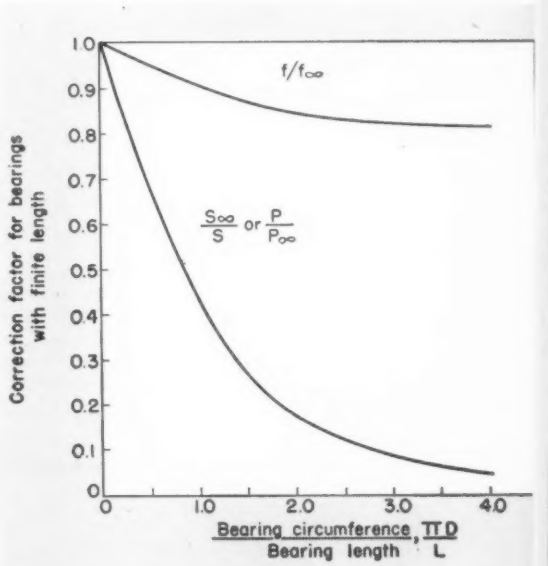

Correction factor for bearings with finite length
f/f∞
S∞/S or P/P∞
1.0 0.9 0.8 0.7 0.6 0.5 0.4 0.3 0.2 0.1 0
0 1.0 2.0 3.0 4.0
Bearing circumference, πD / Bearing length L

Fig. 3—Effect of bearing length on bearing friction, load capacity and Sommerfeld number.

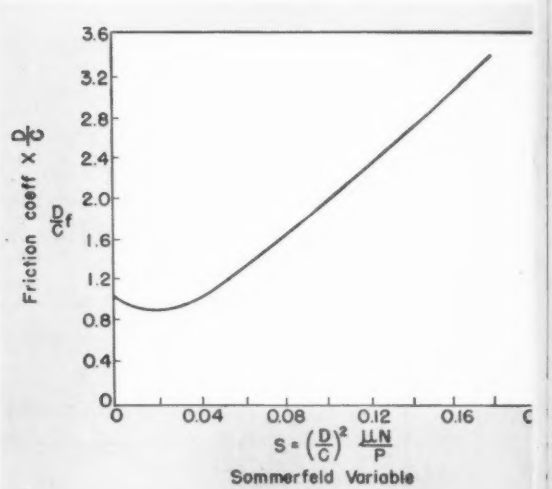


Fig. 4—Relation between Sommerfeld number and friction for a hydrodynamic bearing.

Table I—Formulas for Oil Flow Through Plain Sleeve Bearings

Authority	Formula	Application	What Formula Gives	Comments
Linn	$Q = \frac{16.25P}{\Delta t \times C/3.5}$ where Q = oil flow, gal/min P = power loss, kw Δt = temp rise of oil through bearing, deg F C = specific heat, Btu/gal/deg F	Unidirectionally loaded bearings, lightly loaded (150 psi), minimum of shaft misalignment	Amount of oil required to dissipate friction heat	Used for turbines
Dennison	$Q = 0.5196 \frac{\pi}{6} \frac{P_s a \eta^3}{\mu L} \left(1 + \frac{3}{2} C^2\right)$ where Q = oil flow, gal/min P_s = inlet pressure, psi a = shaft, radius, in. η = radial clearance, in. μ = abs. viscosity, lb-in.-sec $= \frac{Z}{6{,}900{,}000}$ L = 1/2 bearing length, in. C = eccentricity ratio $= \frac{\text{journal eccentricity}}{\text{radial clearance}}$	Dynamically loaded engine bearings, high load and speed	Amount of oil capable of flowing through bearing	
	$Q = 0.26 \frac{S}{\gamma \Delta T}$ γ = heat capacity in in. lb/cu in./deg F (145 for most oils) ΔT = temp rise, deg F Q = oil flow, gal/min	Dynamically loaded engine bearings, high load and speed	Amount of oil required with finite temp rise of oil flowing through bearing	
Kingsbury	$Q = \frac{42.42 \text{ H.P.}}{\Delta t \, \delta}$ where Q = oil flow, gal/min Δt = temp rise, deg F δ = specific heat at mean oil temp over bearings, Btu/gal/deg F	Dynamically loaded engine bearings, high load and speed	Amount of oil required with finite temp rise of oil flowing through bearing	Gives results generally lower than Dennison's formula
Cleveland Graphite Bronze	$Q = \pi D \frac{(C + 0.0005)}{2} \times 2 \times 30$ where Q = oil flow, gpm D = shaft dia, in. C = min diametral clearance, in.	Engine bearings with average dia of 3 in. and shaft speed of 2,500 rpm	Oil flow for adequate cooling and successful performance	Empirical, gives results in close agreement with Dennison's formula
General* Motors	$Q = \frac{\pi}{6} \left(1 + 3/2 \, \eta^2\right) \frac{(P_s R C^3)}{\mu L}$ where Q = oil flow, in.^{3}/sec η = eccentricity, in. P_s = oil supply pressure, psi R = journal radius, in. C = bearing clearance, in. μ = oil viscosity, reynes $= \frac{Z}{6.9 \times 10^6}$ L = 1/2 bearing length	Engine bearings	Amount of oil capable of flowing through bearing	Used for automotive bearings and aircraft engine bearing design
M. C. Shaw	$Q = \frac{C^3 P_o}{3 \mu} \left(1 + 3/2 \, \eta^2\right) \tan^{-1} \frac{(2\pi R)}{L}$ where L = bearing length, in. $\text{Tan}^{-1} \frac{(2\pi R)}{L}$ is in radians Rest of symbols same as in General Motors formula	Engine bearings	Amount of oil capable of flowing through bearing	Used for aircraft engine design

* Used at G. M. and taken from other authorities

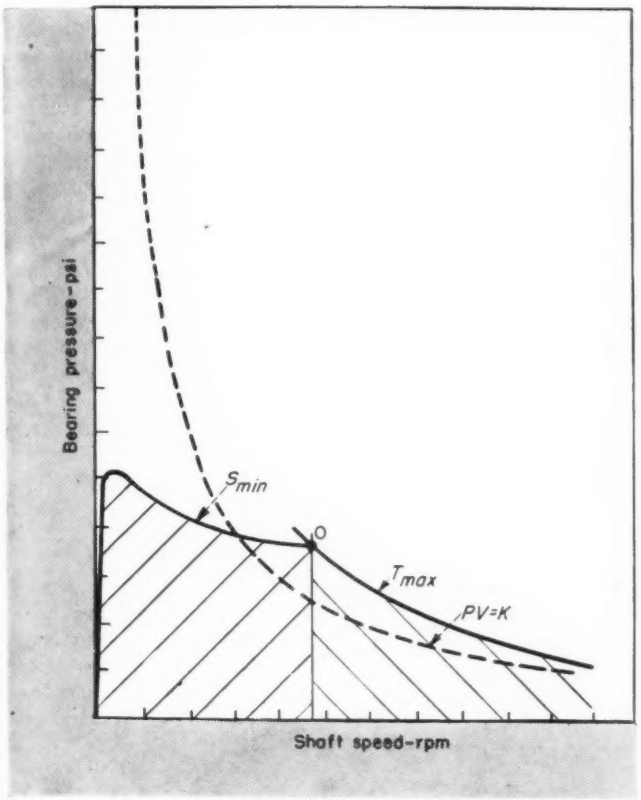


Fig. 5—Typical load speed curve for a hydrodynamic bearing. Note the danger of basing design on PV factor alone.

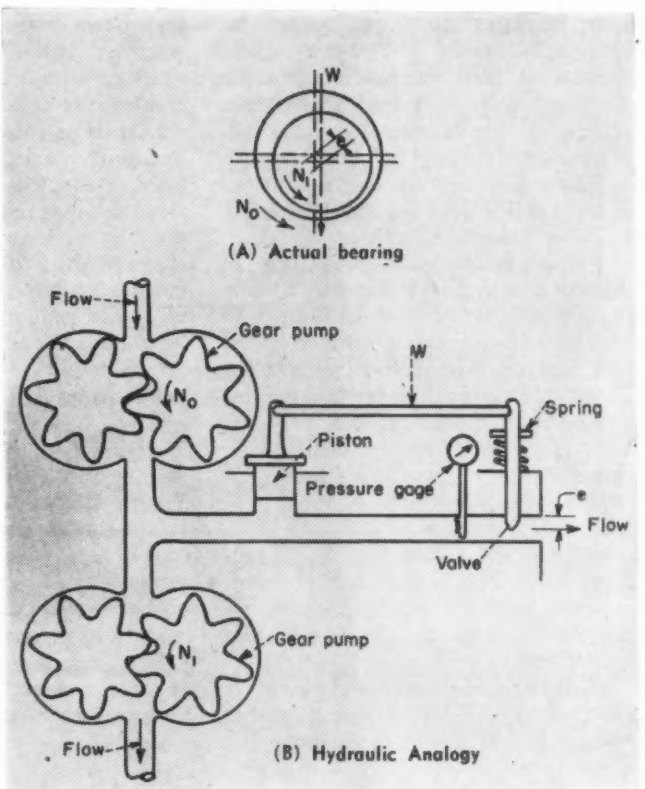


Fig. 6—Hydraulic analogy for hydrodynamic bearing with rotating load. Rotating journal pumps oil around clearance space.

relation among P, N and μ_o obtained. Or if T_{max} is assumed, μ, F and S may be eliminated and a second relation obtained among P, N, μ_o.

As stated, the complete solution for a given bearing requires specific data on the heat dissipating capacity of the bearing and lubricant. For new designs, this information may be difficult to obtain, except by test or by reference to similar installations.

Lubricant Viscosity

The viscosity of commercial oils is expressed in Saybolt Universal seconds, the time for a standard volume of liquid to flow through a short tube of a standard Saybolt instrument. But in hydrodynamic bearing equations, viscosity must be expressed in terms of force, length and time. In the cgs system, the unit of viscosity is the poise, whereas in the English system the unit is the reyn. The relationship between reyn and poise is

$$\underset{\text{(reyn)}}{\mu_o} = 69{,}000 \underset{\text{(poise)}}{Z}$$

To convert from Saybolt Universal seconds to absolute viscosity expressed in cgs units or in English units; the Saybolt reading, (t_s), taken at a specific temperature (usually 100 F or 210 F) is converted to kinematic viscosity Z_k, where

$$Z_k = 0.22t_s - 180/t_s$$

and then to absolute viscosity by

$$Z = \gamma Z_k$$

where γ is the specific gravity of the lubricant at the Saybolt reading temperature.

For other temperatures, Z may be gotten by using the corrected value for γ corresponding to the new temperature. Charts and data defining the various oils used in bearing lubrication are available from the oil companies.

Dynamically Loaded Bearing

The load capacity of dynamically loaded hydrodynamic bearings cannot be computed by the use of the foregoing hydrodynamic and thermodynamic equations. In engine bearings and others, in addition to the film pressure caused by the pumping action of the journal in the bearing, additional pressure is produced by the relative motion of the surfaces towards and away from each other, which takes place when the load rapidly changes in direction and magnitude.

Burwell and Underwood have attempted to refine the basic hydrodynamic formulas to account for the effect of suddenly applied loads on bearing load capacity.

Most dynamically loaded bearings have higher load capacity than statically loaded bearings. The increase in load capacity results from the higher

Table II—Relative Load Capacity for Hydrodynamic Bearings with Dynamic Loading

N_p/N_j	0	½	1	2	3
Constant rotating load	1	0	1	3	5
Sinusoidal alternating load	1	0	1.74	4.72	7.90
Square wave alternating load	1	0	1.25	3.27	5.24

Where: N_p = average frequency of rotation or oscillation of load in revolutions or cycles per unit time
N_j = average angular velocity of journal in revolutions per unit time, a constant

film pressures developed when the bearing is loaded (1), by a rapidly fluctuating load (as in a piston-pin bushing) (2), by a load that rotates relative to the bearing (as in a connecting-rod bearing).

In the first type of loading, a suddenly applied load of short duration tends to squeeze oil out of the clearance space in the loaded area. But the oil acts as a hydraulic damper, absorbing the peak load before oil film thickness is reduced enough to result in film rupture. The momentary increase in film pressure accounts for the increased bearing load capacity.

In the second type of loading, increase in film pressure with load rotation is more difficult to define physically.

Professor M. C. Shaw of M.I.T. has prepared a hydraulic analogy useful in explaining the reason for the increased load capacity of a dynamically loaded bearing.

A dynamically loaded bearing is shown in Fig. 6(A). For the bearing shown, the load is assumed to be rotating, but since relative motions only are significant, the sketch shows the load as constant in direction and the bearing rotating instead. In the hydraulic analogy, Fig. 6(B), the bearing and journal surfaces are considered as gear pumps. These, running at rotation speeds N_0 and N_1, corresponding to the bearing and journal speeds, supply a given volume of oil to the common outlet. In the outlet is a piston and a spring loaded valve (corresponding to the journal and the point of minimum film thickness respectively) which controls fluid flow. The pipe, *P*, in the analogy corresponds to the space between the journal and bearing surfaces in the actual bearing. If the speed of either pump is increased, the flow through the valve is increased and the pressure to the left of the valve increased (and hence bearing load capacity is increased).

Similarly, if one pump slows down or reverses direction, flow and pressure in the common outlet is decreased. Also if load is increased slowly, valve setting, *e*, increases, increasing fluid pressure but decreasing the clearance between bearing and journal. If a sudden load is applied, the action is the same, but in addition, the piston applies a sudden load to the fluid in the outlet line, momentarily increasing pressure over the steady flow pressure.

The action in the actual bearing is the same. If direction of rotation of load and journal are the same, the sum of their speeds is a measure of bearing load capacity and if the applied load on the bearing fluctuates, load capacity is increased further.

Burwell has developed factors for dynamically loaded bearings loaded in various ways. These are shown in Table II.

Underwood lists load capacities for geometrically identical bearings loaded under various conditions, which agree with those shown in the table. It is interesting to note that for one loading condition, when the journal rpm

Table III—Grooves in Unloaded Portion of Bearing: 0.0025 in. Clearance

Oil Supply	Diagram Fig. 8	3,000 rpm				4,000 rpm				
		Bearing Temperature, deg C	Viscosity, centipoises	Coefficient of Friction	Oil Flow, fluid oz per min	Bearing Temperature, deg C	Viscosity, centipoises	Coefficient of Friction	Oil Flow, fluid oz per min	
Bottom hole (no groove)	(a)	106	13.8	0.0014	6	118	10.0	0.0013	7	
1 in. axial groove at bottom	(b)	100	16.1	0.0018	13	109	12.7	0.0019	16	
1½ in. axial groove at bottom	(c)	98	17.0	0.0019	18½	106	13.8	0.0021	23	
90 deg circumferential groove at bottom	(d)	100	16.1	0.0017	13½	110	12.3	0.0018	16½	
180 deg circumferential groove at bottom	(e)	102	15.2	0.0016	11½	110	12.3	0.0017	14½	
270 deg circumferential groove at bottom	(f)	107	13.5	0.0016	8	115	10.9	0.0017	10½	
360 deg circumferential groove	(g)	115	10.9	0.0020	8½	129	7.7	0.0021	10	
360 deg circumferential groove, with ½ in. narrower bush.	(h)	118	10.0	0.0026	11½	133	7.1	0.0027	13	
Hole at 90 deg., inlet side	(i)	115	10.9	0.0012	2½	124	8.7	0.0012	3½	
Hole at 90 deg, outlet side	(i) reversed	128	7.9	0.0010	1½	138	6.4	0.0010	2	
1 in. axial groove at 90 deg, inlet side	(j)	107	13.5	0.0013	6	113	11.5	0.0015	8½	
1 in. axial groove at 90 deg, outlet side	(j) reversed	108	13.0	0.0012	5	117	10.2	0.0014	7½	
1½ in. axial groove at 90 deg, inlet side	(k)	104	14.3	0.0014	11	115	10.9	0.0015	14	
1½ in. axial groove at 90 deg, outlet side	(k) reversed	106	13.8	0.0013	9	117	10.2	0.0014	12	
groove	(l)	98	17.0	0.0019	19½	106	13.8	0.0021	23½	
groove reversed	(l) reversed	97	17.6	0.0019	24	104	14.3	0.0021	29	
groove	(m)	98	17.0	0.0019	19½	106	13.8	0.0022	24	
groove reversed	(m) reversed	98	17.0	0.0020	25	106	13.8	0.0023	28½	
Effect of oil inlet temperature:										*Oil temperature, deg C*
360 deg circumferential groove	(g)	109	12.7	0.0023	5½	...	....	.....	....	31
360 deg circumferential groove	(g)	115	10.9	0.0020	8½	...	....	.....	....	70
Effect of oil inlet pressure[d]										*Oil pressure, psi*
1½ in. axial groove at bottom	(e)	99½	16.5	0.0020	19	...	....	.....	....	100
1½ in. axial groove at bottom	(e)	100½	15.8	0.0019	18½	...	....	.....	....	80
1½ in. axial groove at bottom	(e)	101	15.6	0.0019	17½	...	....	.....	....	60
1½ in. axial groove at bottom	(e)	101½	15.4	0.0018	16	...	....	.....	....	40
1½ in. axial groove at bottom	(e)	....	....	.....	....	104½	14.1	0.0020	27	120
1½ in. axial groove at bottom	(e)	....	....	.....	....	106	13.8	0.0020	23	80
1½ in. axial groove at bottom	(e)	....	....	.....	....	107½	13.2	0.0020	21	40
Effect of speed:										*Speed, rpm*
360 deg circumferential groove	(g)	104	14.3	0.0020	6½	....	....	.....	....	2,000
360 deg circumferential groove	(g)	111	12.0	0.0020	7½	....	....	.....	....	2,500
360 deg circumferential groove	(g)	113	11.5	.....	7½	....	....	.....	....	2,720
360 deg circumferential groove	(g)	129	7.7	0.0021	10	....	....	.....	....	4,000
360 deg circumferential groove; ½ in. narrower bush.	(h)	110	12.3	0.0025	10½	....	....	.....	....	2,500
360 deg circumferential groove; ½ in. narrower bush.	(h)	133	7.1	0.0027	13	....	....	.....	....	4,000

Table IV—Grooves in Unloaded Portion of Bearing: 0.0041 in. Clearance (Bearing Case Off)

Oil Supply	Diagram Fig. 8	3,000 rpm				
		Bearing Temperature deg C	Viscosity, centipoises	Coefficient of Friction	Oil Flow, fluid oz per min	
Bottom hole	(a)	98	17.0	0.0015	15½	
1½ in. axial groove at bottom	(c)	92	20.9	0.0017	40	
90 deg circumferential groove at bottom	(d)	92	20.9	0.0017	32	
Hole at 90 deg, inlet side	(i)	107	13.5	0.0012	5½	
Hole at 90 deg, outlet side	(i) reversed	111	12.0	0.0011	4	
1½ in. axial groove at 90 deg, inlet side	(k)	99	16.7	0.0015	17	
1½ in. axial groove at 90 deg, outlet side	(k) reversed	97	17.6	0.0015	17	
Effect of oil inlet temperature:				*3,000 rpm*		*Oil Temperature deg C*
Bottom hole	(a)	91	21.6	.....	8	40
Buttom hole	(a)	92	20.9	0.0018	9	46
Bottom hole	(a)	94	19.5	0.0017	11½	55
Bottom hole	(a)	98	17.0	0.0015	15½	70
				2,500 rpm		
Bottom hole	(a)	83	28.1	0.0017	7½	41
Bottom hole	(a)	85	26.2	0.0016	....	50
Bottom hole	(a)	92	20.9	0.0014	14	70
Effect of oil inlet pressure:				*2,500 rpm*		*Oil Pressure, psi*
Bottom hole	(a)	91	21.6	.....	16½	101
Bottom hole	(a)	92	20.9	0.0014	14	80
Bottom hole	(a)	93	20.1	.....	13	60
Bottom hole	(a)	94	19.5	.....	11	40

is twice load rotation frequency, bearing load capacity is zero. This condition can occur in internal combustion engines; but only for a very short time compared to the complete load cycle. If it occurs frequently, an antifriction bearing may be used.

As shown in Table II, Burwell's load capacity factors apply only to the specified loading conditions. In automotive engines, both load and speed vary in such a way that no mathematical solution for load capacity is possible. Underwood has made studies of actual connecting rods in a diesel engine and has plotted bearing load capacity versus crank angle in terms of the average load and speed conditions that exist during each successive ten degree position of the crankshaft. The results for one study are shown in Fig. 7. As shown, load capacity is zero several times during one revolution of the crankshaft.

The results of these investigations are valuable chiefly because they indicate the need for more study of this problem and because they explain why the load capacity of identical bearings, lubricated with oil of the same viscosity, may vary with changes in operating condition.

Oil Grooves

Few reliable data exist for oil groove design, variation of oil flow with change in groove design, and effect of oil inlet pressure on bearing performance. In the discussion on oil flow, page 152, the oil flow formulas shown apply only to bearings with a circumferential groove. Past experience indicates that bearing performance is materially affected by changes in oil inlet locations although a bearing is affected materially by oil grooving.

Engine bearings must be carefully designed for adequate lubrication because of severe operating conditions.

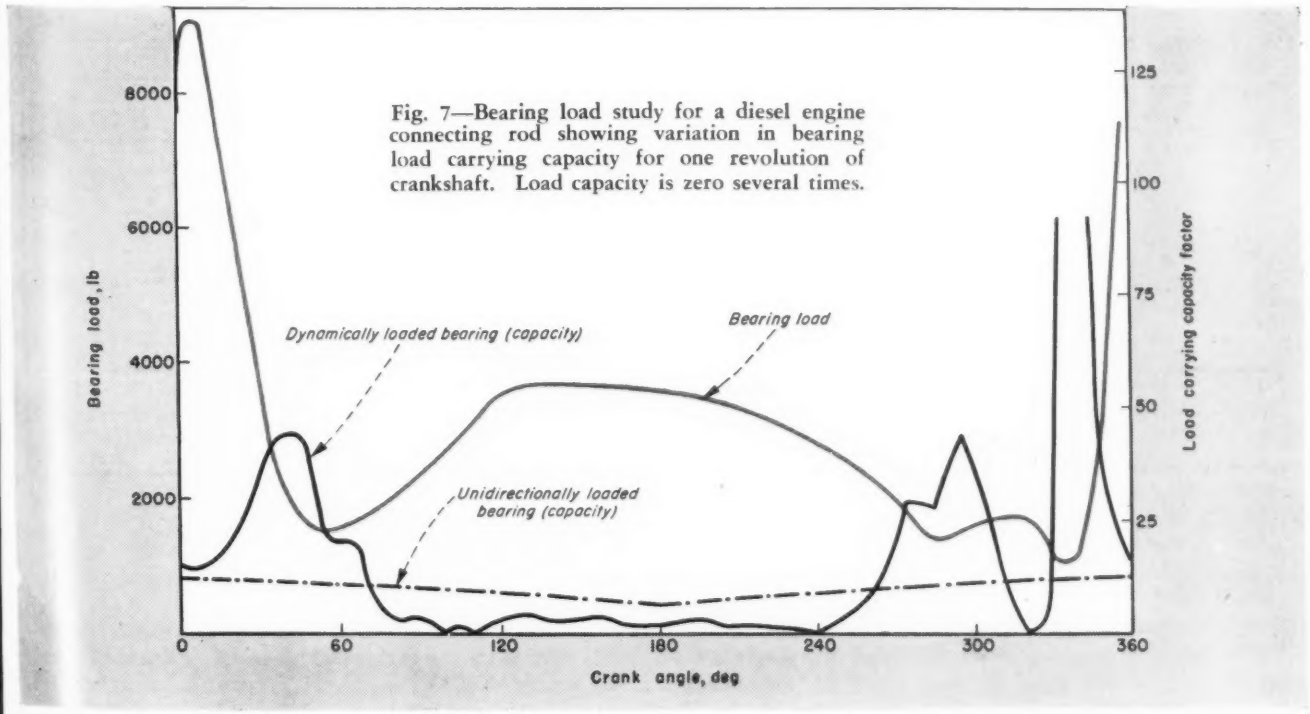


Fig. 7—Bearing load study for a diesel engine connecting rod showing variation in bearing load carrying capacity for one revolution of crankshaft. Load capacity is zero several times.

Table V—Grooves in Loaded Portion of Bearing; 2,500 rpm

Hole or Groove at Crown of Bush	Diagram Fig. 9	Top Hole Open					Top Hole Closed				
		Bearing Temperature, deg C	Viscosity, centipoises	Coefficient of Friction	Oil Flow, fluid oz per min		Bearing Temperature, deg C	Viscosity, centipoises	Coefficient of Friction	Top Pressure, psi	Side Flow fluid oz per min
					Side	Top					
¼ in. dia hole	(a)	110	12.2	0.0018	5½	7/8	107	13.4	0.0014	2,980	5½
¼ in. hole +0.3 in. groove extending hole on inlet side of film	(b)	111	12.0	0.0019	6	1½	104	14.4	0.0015	2,400	5½
¼ in. hole +0.6 in. groove, do	(c)	110	12.2	0.0017	4¾	2¾	108	13.0	0.0015	1,420	6½
¼ in. hole +0.9 in. groove, do	(d)	108	13.0	0.0019	4¼	3¾	106	13.8	0.0017	930	7
¼ in. hole +1.2 in. groove, do	(e)	110	12.2	0.0019	2½	5½	110	12.2	0.0018	500	7¼
¼ in. hole +groove to 90 deg do	(f)	110	12.2	0.0019	1¾	6½	110	12.2	0.0017	314	7¾
do.+do.+⅛ in. groove, extension on outlet side	(g)	108	13.0	0.0020	2¾	6	108	13.0	0.0018	356	7¾
do.+do.+1 in. groove, do	(h)	109	12.7	0.0021	2½	5½	110	12.2	0.0018	330	7
180 deg groove	(i)	108	13.0	0.0019	2½	5¾	110	12.2	0.0019	260	6½
360 deg groove	(j)	104	14.4	——	4¾	——	110	12.2	0.0019	——	7¼
½ in. narrower bush (1¼ in. wide) —360 deg groove	(k)	100	16.1	0.0030	6¾	—	104	14.4	0.0026	——	9½
¼ in. narrower bush (⅞ in. wide) —360 deg groove	(l)	97	17.7	0.0052	11	—	101	15.7	0.0053	——	15¾
½ in. narrower bush (1¼ in. wide) —360 deg groove; 2,100 psi	(k)	105	14.1	0.0028	8	—	111	12.0	0.0026	——	11
¼ in. narrower bush (⅞ in. wide) —360 deg groove; 3,000 psi	(l)	107	13.4	0.0037	15¼	—	110	12.2	0.0035	——	19
(g) above with 80 psi back pressure	(g)	110	12.2	0.0020	2¼	5¼	——	——	——	——	——
Effect of speed:											
180 deg. groove; 2,500 rpm	(i)	108	13.0	0.0020	2¾	4¾	——	——	——	——	——
180 deg groove; 3,000 rpm	(i)	115	10.8	——	2¾	6¾	——	——	——	——	——
180 deg groove; 3,500 rpm	(i)	122	9.1	——	2¾	7½	——	——	——	——	——
180 deg groove; 4,000 rpm	(i)	128	7.9	0.0021	2¾	8½	——	——	——	——	——
Direction of rotation reversed											
¼ in. dia hole	(a) reversed	110	12.2	0.0018	5¼	1	106	13.8	0.0015	3,200	5¼
¼ in. hole +0.3 in. groove extending hole on outlet side of film	(b) reversed	110	12.2	0.0019	5¼	1¼	106	13.8	0.0015	3,250	5½
¼ in. hole +0.6 in. groove, do	(c) reversed	107	13.4	0.0017	5¼	1⅜	105	14.0	0.0015	2,900	5¾
1 in. hole +0.9 in. groove, do	(d) reversed	106	13.8	0.0019	5¼	1⅜	104	14.4	0.0017	2,130	5¾
¼ in. hole +1.2 in. groove, do	(e) reversed	109	12.7	0.0019	5¼	1⅛	106	13.8	0.0018	840	5¼
¼ in. hole+groove to 90 deg do	(f) reversed	107	13.4	0.0019	5¼	¾	110	12.2	0.0017	90	6
do.+do.+⅛ in. groove extension on inlet side	(g) reversed	108	13.0	0.0020	4¾	2¼	110	12.2	0.0018	262	5¼
do.+do.+1 in. groove extension on inlet side	(h) reversed	110	12.2	0.0021	3⅛	4	111	12.0	0.0019	266	5¼
180 deg groove	(i) reversed	108	13.0	0.0019	2¼	5	110	12.2	0.0019	246	6½
(g) above with 80 psi back pressure	(g) reversed	111	12.0	0.0020	5	1¼	—	—	——	——	——

Table VI—Grooves in Loaded Portion of Bearing—Effect of Oil Supply Conditions; 2,500 rpm

Groove	Oil Supply Conditions		Top Hole Open					Top Hole Closed				
	Pressure, psi	Temperature, deg C	Bearing Temperature, deg C	Viscosity, centipoises	Coefficient of Friction	Oil Flow, fluid oz per min		Bearing Temperature, deg C	Viscosity, centipoises	Coefficient of Friction	Top Pressure psi	Side Flow fluid oz per min.
						Side	Top					
Groove from 90 deg on inlet side to ½ in. beyond top hole on outlet side (g of Fig. 9)	80	70	110	12.2	0.0021	1¾	6½	112	11.8	0.0021	256	8
	40	70	113	11.5	0.0020	1	6	113	11.5	0.0020	235	6½
	80	100	...	....	.....	...	...	125	8.5	0.0019	204	12¼
Do., with 80 psi back pressure at top hole.	80	70	109	12.7	0.0022	2¾	5	...	....	.....	...	...
Groove from 90 deg on outlet side to ½ in. beyond top hole on inlet side (g of Fig. 9 reversed)	80	70	110	12.2	0.0021	5¾	1⅛	110	12.2	0.0022	140	6¼
	40	70	113	11.5	0.0022	4½	1	115	10.8	0.0020	100	5
	80	100	...	....	.....	...	...	122	9.2	0.0019	52	9½
Do., with 80 psi back pressure at top hole.	80	70	110	12.2	0.0022	6¼	½	...	....	.....	...	...

Clayton has investigated the effect of oil grooving on the performance of this type of bearing. His work is comprehensive, and, is possibly the best reference as a guide to groove design.

Clayton tested a series of 2 in. diameter engine bearings made of phosphor-bronze and of copper-lead operating on nickel chromium steel journals. In these, oil inlet location and groove design were systematically changed. Bearing pressure was 1,500 psi and journal speeds were 2,000, 3,000 and 4,000 rpm. The object of the tests was to study changes in oil groove design and effect on bearing performance caused by locating the oil inlet (1) in the unloaded portion of the bearing, and (2) in the loaded portion of the bearing.

In Figs. 8 and 9 are shown the designs tested, and in Tables III, IV, V and VI, the test results are recorded. These indicate that for oil

hole inlets and grooves located in the unloaded portion of the bearing

1. Oil flow through single hole feeds is markedly influenced by clearance at that point.
2. Oil flow is greater if the oil hole feeds into grooves.
3. Circumferential grooves extending more than 270 deg do not increase oil flow and actually decrease safety.
4. Inclined grooves increase oil flow.

Also, if oil inlet and grooves are located in the loaded portion of the bearing, the oil flow is markedly reduced.

Oil Groove Design—Low-Speed Medium-Load Bearings

Special oil grooves may be used for low-speed, medium-load bearings. Recommended dimensions for grooving in these vary greatly depending on the source. However, as in the design of oil grooving for high-speed, highly-loaded bearings, some established data that has been found to work fairly well in practice is better than resorting to guesses.

The following groove dimensions have been used with some success for the low-speed, medium-load bearings.

Bearing dia, in.	Groove width*, in.	Groove depth, in.
0.5 to 1	9/64	3/64
1 to 1.5	3/16	1/16
1.5 to 2.5	9/32	3/32
2.5 to 4	3/8	1/8
4 to 5.5	9/16	3/16

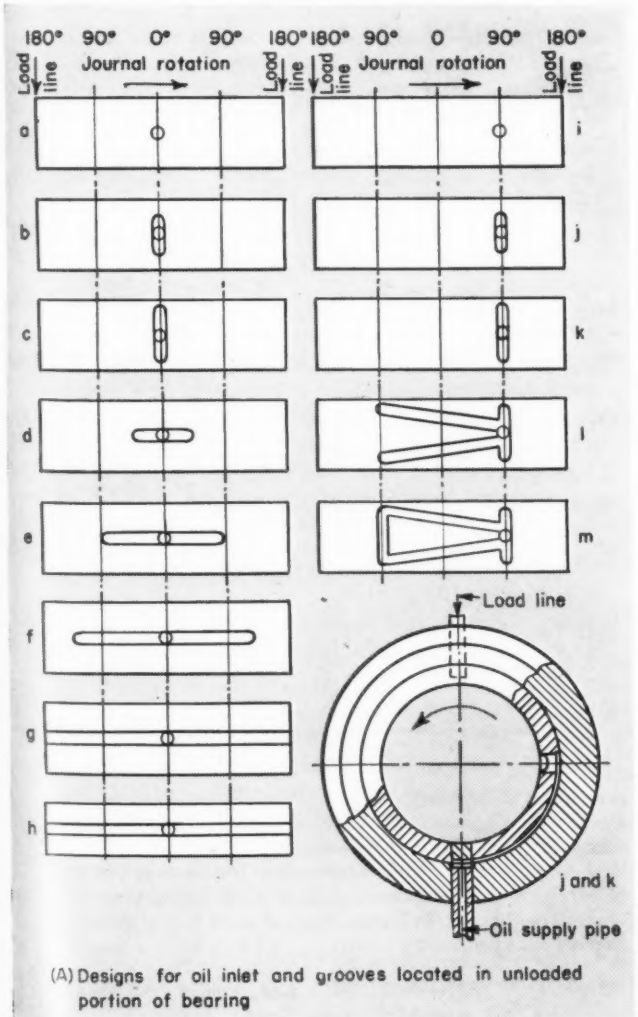
* Parallel to bearing axis.

Concluding Notes on Plain Sleeve Bearings

The Product Engineering survey on plain sleeve bearings indicated that although there are much data relating to bearing material and design, bearing design is still largely an art rather than a science. The data gathered together and presented in this special section on plain sleeve bearings came from many sources and reflects current design practice. Obviously much development work remains to be done to provide designers with the kind of engineering data that will allow them to design bearings in a more rigorous manner.

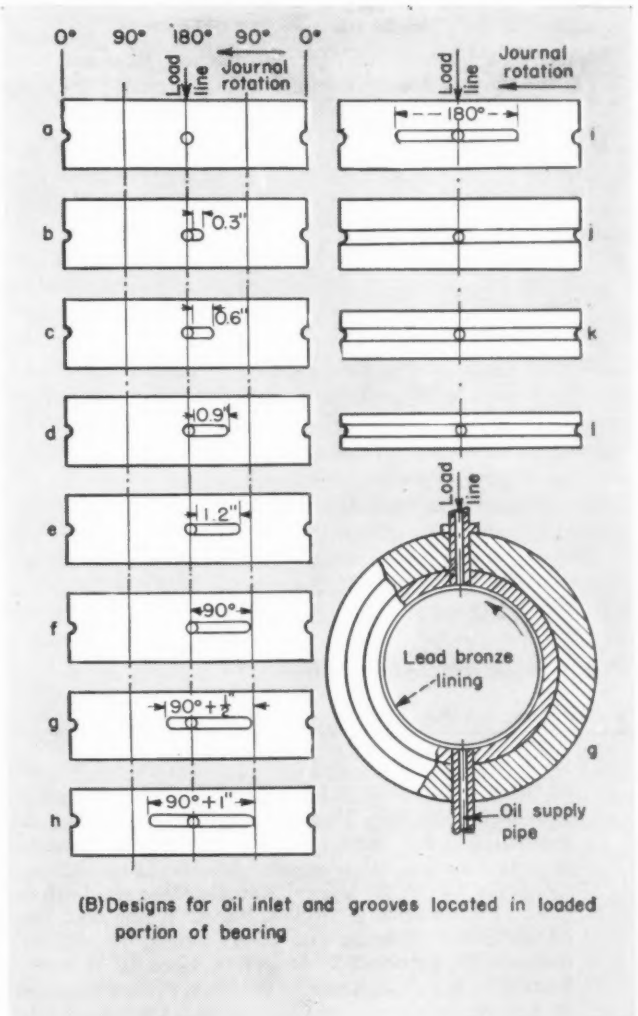
More should be known about the bearing qualities of metals, the effect of operating variables, and the extension of theoretical methods for practical application. Much development work remains to be done on the investigation of the effects of dynamic loading on the load carrying capacity of bearings, on the effects of oil hole and groove design on bearing performance, and the development of rational methods for selecting performance criteria.

It appears that much unpublished data on bearing materials and design can be brought to light through greater cooperation between the bearing manufacturers and bearing users. In this connection the technical societies afford an excellent meeting ground.



(A) Designs for oil inlet and grooves located in unloaded portion of bearing

Fig. 8—Variations in oil hole locations and oil groove designs for engine bearings—located in unloaded portion of bearing.



(B) Designs for oil inlet and grooves located in loaded portion of bearing

Fig. 9—Variations in oil hole locations and oil groove designs for engine bearings—located in loaded portion of bearing.

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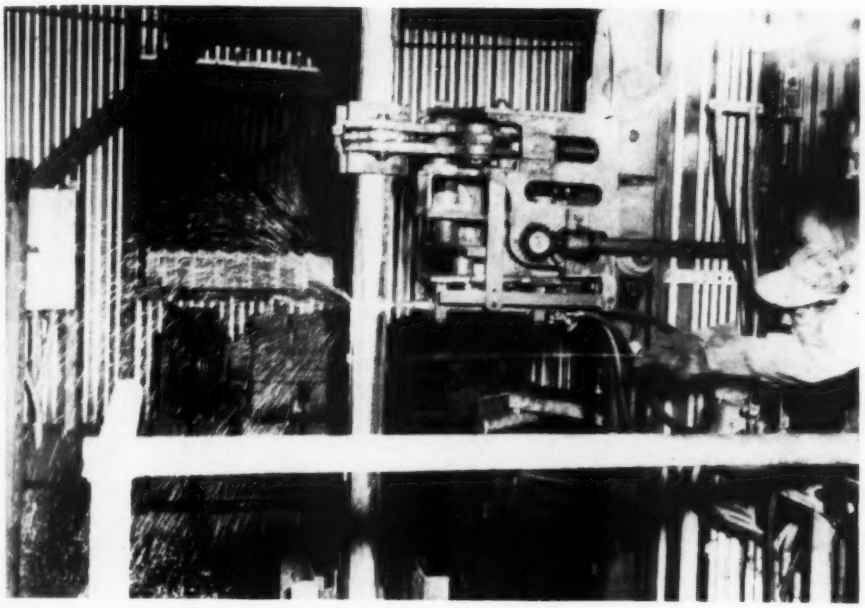
PRODUCT ENGINEERING thanks the following individuals for the help they gave and the data they contributed for this special section . . . L. E. Cardwell, Micarta Division, Westinghouse Electric Corp; B. Carp, Morganite Inc; R. W. Dayton, Battelle Memorial Institute; C. A. Drach, Lucian Q. Moffitt, Inc; C. R. Johnson, Paramount Oilless Bearing Co; C. V. Lynch, Jr., Franklin Institute Laboratories; S. A. McKee, National Bureau of Standards; J. Palsulich and R. W. Blair, Project Engineers, Wright Aeronautical Corp; E. A. Ryder, Consulting Engineer, Pratt & Whitney Aircraft; M. C. Shaw, Professor in Mechanical Engineering, M.I.T.; I. C. Sleight, Chief Metallurgist, P. R. Mallory and Co., Inc; W. A. Smith, The B. F. Goodrich Co; G. J. Talbourdet, Engineer in Charge of Mechanical Mechanisms, United Shoe Machinery Co; W. R. Toeplitz, V. P. Engineering, Bound Brook Oil-less Bearing Company . . . .

PRODUCT ENGINEERING also thanks the many companies that responded to the bearing survey.

TECHNICAL NEWS



CONTINUOUSLY CAST STEEL BILLET emerges from machine and is cut to specified length by an oxy-acetylene torch that moves vertically down at the casting speed.

Simple Machine Casts Steel Continuously

SEMI-FINISHED STEEL SHAPES have been made directly from the liquid phase in one simple relatively inexpensive operation and a century-old dream has been realized. The machine, a joint development of Republic Steel Corp. and The Babcock & Wilcox Tube Co., is in pilot plant operation at Beaver Falls, Pa.

In March, 45 tons of 0.15 C steel, continuously cast 6 in. rounds, were shipped to a commercial user. More recently, continuously cast slabs have been rolled into strip. This strip has been converted to electric-welded boiler tubes meeting all physical requirements for the company's regular product.

Continuous casting of semi-finished steel will introduce possibilities of utmost importance to the steel industry and users of steel products. Increased productivity and decentralized production brought about by simplification of apparatus and increasing final yield of steel from the original melt will be major advantages. The new method and apparatus are a timely response to economic pressure demanding local delivery (PRODUCT ENGINEERING, September 1948, page 161—Cement Institute Decision). Interest in the possibilities of continuously cast semi-finished steel is enhanced by current uneasiness over high break-even points and the doubled and tripled cost of new productive facilities.

Greatest single advantage of this process, according to the developers, is that it removes from the process of steelmaking the most massive and expensive parts of such plants, such as equipment for ingots, soaking pits and the blooming mill. Continuous casting produces semi-finished sections ready for secondary mills with the result that not only is capital cost for a given output greatly reduced, but also maintenance and operational costs are lowered. The work accomplished to date makes it clear that for production of relatively small quantities of steel and for decentralization of steel production facilities, this new development fills requirements for low capital and low operational costs.

The Beaver Falls machine casts two or three times a week, handling both carbon and alloy steels. Each continuous cast products 5,000 lb of steel billets. A section of about 30 sq in. is cast at a rate of 400 lb per minute. This is equivalent to a linear forming rate of 57½ in. per minute. It is expected that continued work on the section shape will increase the forming rate. A mold is now being considered that will have a cross-section area of 45 sq in. Results have indicated that ovals of special proportions are the most practicable shapes to continuously cast. The oval should be as flat as can be handled effectively in the rolling mill. The oval shapes can be rolled into rounds or flats.

The casting machine is located at the top of a 75 ft tower. Steel is delivered from the company's regular electric furnaces to the top of the tower in transfer ladles. An inductively heated holding and pouring ladle us used to retain the metal before a cast is made. This arrangement was found to be the best for an experimental unit relying on an intermittent metal supply.

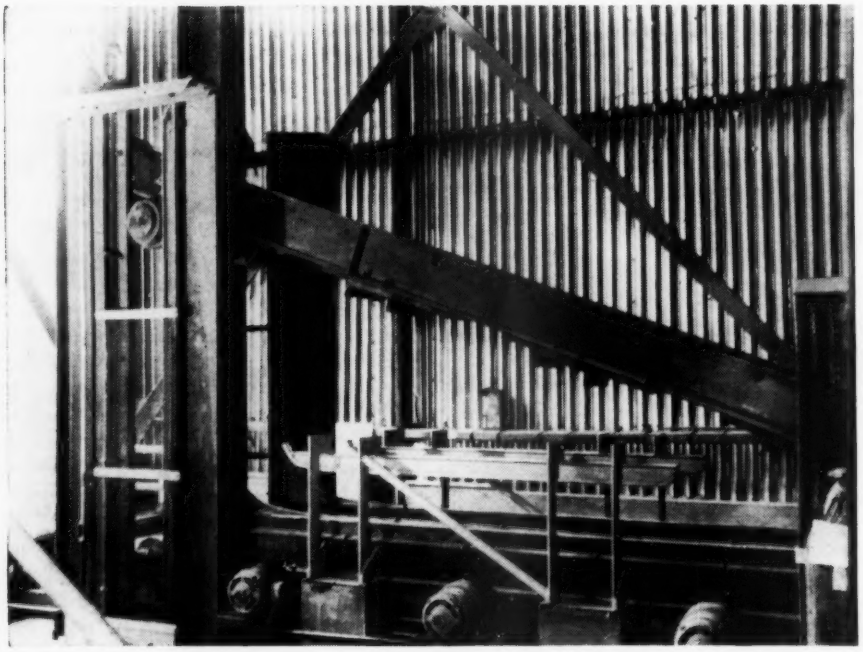
A mirror arrangement has been set up so that the surface of the steel can be seen. The steel must have a strong negative meniscus so that it does not wet the mold, and must be clean at all times. The mold is not lubricated but a small amount of hydrocarbon is introduced into the mold to combine with and eliminate free oxygen above the metal pool. This is necessary because oxides in the steel promote wetting of the mold.

Since continuous casting requires a slag-free steel, the molten metal is poured from the holding furnace into a tundish, designed to strain out the slag. In the experimental unit, a preheated tundish is used, but an electrically heated one would be used in actual production.

Shortly after the shell of the billet has been formed, the steel contracts and in so doing the billet shrinks from the mold. The metal contacts the surface of the mold for only a few inches and only in this short distance can it lose heat to the mold by direct contact. Proper cooling of the mold is essential and its importance increases as casting rate increases. Circulating water at a rate of 500 gpm is used to remove heat absorbed by the mold surface.

Mold materials can be of different metals. Steel, $\frac{1}{16}$ in. thick; copper, $\frac{1}{4}$ in. thick, and brass $\frac{3}{16}$ in. thick have been used for molds. These thicknesses give approximately the same rate of heat transfer for the molds. Brass was found to have definite structural and fabricational advantages and stainless steel was found to be impractical because it promotes wetting.

Below the mold, the casting passes through an insulated chamber that slows the rate of cooling. Below this chamber are the pinch rolls that reg-



CRADLE LOWERS BILLETS, up to 35 ft long, cut from continuous casting to horizontal position on roller conveyor then resumes upright position to receive next section.

ulate the downward speed of the billet. The casting then passes an oxyacetylene torch that travels at billet speed until the casting is cut to specified length. The lengths can be as long as 35 ft and are lowered to horizontal position by a cradle arrangement.

Continuous castings meet established criteria for sound steelmaking practice and have certain advantages. The process allows faster cooling, which results in a fine and uniform crystalline structure with little segregation. It offers less opportunity for dirt or other foreign material to enter the casting. The surface of the semi-finished billet will be freer of checking and scabs and its interior will be free of entrapped slag.

The heat can be withdrawn from the billet in any pattern desired and the machine is designed so that the steel in the mold solidifies progressively from the bottom to the top. Although the mold is uniform in cross-section, it approximates in behavior the big-end-up mold, the type used for the best quality steels made by conventional processes. Continuously cast ingots cool rapidly and therefore tend toward a fine grain rather than columnar crystal structure. The billets can be produced without taper.

All the steel produced by this process has shown physical properties as good as steel made by conventional processes, but not enough metallurgical investigation has been completed to indicate whether the physical properties are superior. However, it has been definitely shown that equivalent steels can be produced more cheaply by this continuous casting process.

Wind Tunnels Blow Hot and Cold To Study Oil Cooler

THE FIRST BRAZED and welded aluminum jet plane oil cooler has recently been designed and tested in the "hot" and "cold" wind tunnels at the Clifford Manufacturing Div. of Standard-Thomson Corp. Weight-saving of 200 to 600 lb over copper oil coolers has been attained with a design that preheats jet fuel. There are from 600 to 6,000 aluminum tubes, 0.006 in. thick, in each cooler, depending on its capacity rating.

The research laboratory consists of two wind tunnels; a hot run tunnel that normally operates at 100 F inlet air temperature, and a refrigerated tunnel designed to have a capacity large enough to supply twice rated air to a 24 in. dia oil cooler at —60 F when oil at 225 F is flowing through the cooler.

Both tunnels have high speed fans directly coupled to magnetic clutches and driven by squirrel cage motors. Fan speed can be controlled over an infinite range of values by means of the magnetic clutch. Design capacity of the hot tunnel is 48,000 cfm at 70 F and the capacity of the cold tunnel is 24,000 cfm at the same temperature.

Inlet air temperature in the hot run tunnel is maintained by drawing atmospheric air into the tunnel and mixing it with the air in the tunnel in such proportions as to obtain the desired temperature. Refrigeration plant consists of two centrifugal compressors arranged in series.

All test measurements are read in a central control room and include: Rate of oil flow, inlet oil temperature, outlet oil temperature, inlet air temperature, outlet air temperature, and the air flow in pounds per minute. Because of the complexity of present day military requirements it often takes three days to test a single cooler to make certain that it will be effective at speeds up to 750 miles per hour.

Government Releases Data On Magnetic Metals

GOVERNMENT DISCOVERED DATA was released to industry at a recent Symposium on Magnetic Metals held under auspices of the Naval Ordnance Laboratory. Reason for this departure from usual methods was that NOL hoped industry could make use of the information in such way as to greatly benefit both industry and the armed services. Because of curtailed research appropriations to the military, the government wants industry to carry on full research programs based on as much information as possible.

One of the major releases concerned a new magnetic allow, Permanorm 5000-Z, that may possibly revolutionize the rectification of electric power. It may do away with need for a large number of vacuum tubes.

The alloy, resulting from fusion of equal parts of nickel and iron under special heat-treatment, was first made in 1943 in Germany.

The metal has important application in the construction of mechanical rectifiers where it makes possible conversion of alternating to direct current and gives high efficiency at low voltage. It is also possible to convert direct to alternating current, but this has not been developed extensively.

This material may be used in magnetic amplifiers, where it will give linear response and will have twice the power of those made with standard material it is claimed.

Washington Notes

Synthetic Mica Produced

Climaxing two years of effort, synthetic mica, strategic stockpile mineral used in electronic equipment, is being produced in sufficient quantities for experimental evaluation. The synthetic material is being made on pilot-plant scale under a coordinated research program sponsored by the Office of Naval Research, the Army Signal Corps and the Navy Bureau of Ships.

Production of synthetic mica on a commercial scale will make the United States independent of foreign sources. Although natural mica is found in several parts of this country, and despite a government subsidy of six dollars per pound, only 15 percent of the amount used in the critical war years of 1943 and 1944 was produced domestically. The government has considered the mica supply sufficiently acute to designate needed grades of the mineral as Class A strategic materials under the recently enacted Stockpiling Act.

Inherent in the new mica synthesis is the possibility that further research may reveal ways of directly forming mica components. This would avoid the time consuming and laborious tasks of sorting, grading, splitting and trimming natural mica.

Capehart Committee Sympathetic To Industry on Delivered Prices

Representatives of producers, processors, and leaders in labor and farming will serve as an Advisory Council to Sen. Homer Capehart (R, Ind.) and his committee in a study of basing-point and delivered pricing systems. The council will hear pro and con reaction to recent basing-point decisions by the Supreme Court and the Federal Trade Commission.

The council of businessmen, which will make separate recommendations to the Congressional investigating committee, has started to work. It includes top executives from companies producing cement, firebrick, rubber products, textiles, steel, corn products and chemicals—all products where freight rate is large percentage of delivered price.

There is no question but that the Capehart committee is sympathetic to industry's side of this battle with FTC and the courts on delivered pricing. General counsel for the full committee is associated with a firm that represented the salt producers in a delivered pricing scrap with FTC several years ago. The 40-man Advisory Council itself is made up mainly of spokesmen for business. Senator Capehart has summed up his understanding of the question in a way that indicates he is anxious to lift any burden from the Cement Decision that may affect business adversely.

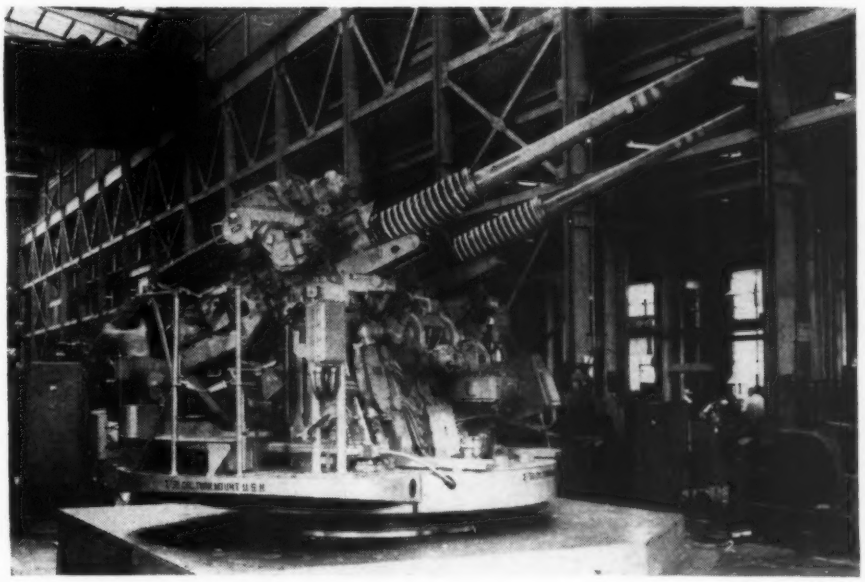
Schedule now calls for the council to hand in its recommendations before the first of the year. The Capehart committee will begin public hearings right after election, and must report, by March 15, 1949, to the Senate.

AEC Forsees Atomic Power In General Use in Two Decades

The General Advisory Committee of the Atomic Energy Commission, long regarded as the most pessimistic government group towards power possibilities, has presented a semi-annual report to the full commission that indicates atomic energy in this country may be as great or greater than present output of coal power "and will operate at lower cost, at least as far as fuel expenditure is concerned." This prediction is based on two assumptions: (1) That the cost of uranium does not rise markedly in the future because of forced resort to low-grade ore; and (2) that the theoretical possibilities of "breeding" atomic fuel can be realized in engineering practice. If these assumptions are in error, atomic power could compete with coal power only in regions where cost of transportation from the mine is a determining factor.

This view of economic prospects for atomic power is the most optimistic that has come from official sources. It puts the official AEC position more nearly in line with private estimates that the ultimate costs of atomic power will fall somewhere between those for hydro and steam.

The air-cooled natural uranium pile at Brookhaven Laboratory on Long Island should be completed early next year, but it will produce little electrical energy. The heat energy from the cooling air will produce less electrical



Spawned by Kamikaze Planes

THIS NEW RAPID FIRE, twin mount 3 in.-0.50 caliber dual purpose gun will fire the famed VT or proximity fuzed ammunition from the decks of many of the major combatant ships of the U. S. Fleet. The mounts will provide effective close-in defense against high speed aircraft and missiles. This gun had its inception off the beaches of Leyte where the Japanese Kamikaze planes constituted a major threat to naval units. It is the result of combined development by the United Shoe Machinery Corp., General Electric Co., the Naval Gun Factory and the Bureau of Ordnance.

According to Rear Admiral A. G. Noble, Chief of the Bureau of Ordnance, these new mounts give from four to six times more fire power than the guns they are replacing. The greatest improvement over the manually operated single three-inch gun is in the method of loading. The new guns are loaded automatically, which is an important advance in gun design.

power than that required for its own cooling system blowers.

Another reactor of more advanced design has been approved for construction at the Argonne Laboratory in Chicago. This will be a fast-neutron pile using enriched fuel and cooled by liquid metal. It will run at a high temperature and will produce power approaching commercial quantities. The advisory committee thinks that reactors, useful for special purposes, should be available within ten years.

Large scale production of price-competitive atomic power would probably require another ten years.

Fuel Crisis Eases

Threatened summer gasoline shortage did not occur this year and the fuel picture is improved over this time last year. However, continued equipment shortages and ever-increasing consumer demand are running improved deliveries a close race.

Outlook for the three basic fuels show: Coal, generally good; fuel oil, considerably better than last year; and natural gas, better but still not bright.

Coal: Except for a shortage in some qualities, and barring transportation tie-up or mine strikes, no one will be seriously lacking in coal this winter. As of July 31, coal stocks in the hands of consumers totaled 58 million tons, enough for 45 days.

Fuel Oil: Production of fuel oil is ten percent higher than last year, but consumption is up almost ten percent. Prospects are better but officials point out there will still be need for consumer conservation. Demand for distillate fuel oil, kerosene, and residual fuel oil shows respective increases, this year over last, of 15, 16 and less than 5 percent. Delivery capacity is also up over last year but more pipe lines and a bigger fleet of tank cars are needed.

Natural Gas: The two sections hit hardest by last year's natural gas shortage were the Appalachian and certain midwest areas. The Appalachian area, including New York, Pennsylvania, Kentucky, West Virginia and Ohio is scheduled to get 20 percent more than last winter; the Midwest will get 15 percent more. Improved pipe line capacity and bigger storage facilities are the reasons for higher deliveries.

Foreign Reports from McGraw-Hill World News

Radio Links Aussie Ranchers To Telephone Network

First radiophone equipment of small size, developed by Electronics Industries Ltd., Melbourne, is adding New South Wales ranchers to the community of telephone subscribers. The set looks much like a 1935 vintage radio receiver.

Subscribers use a standard telephone handset and only three operating controls are necessary. The first is an on-off switch. The second control is a press-to-talk switch that actuates the transmitter and changes the antenna from receiving to transmitting. The tuning dial is set for "day", "night", "summer" and "winter" to obtain optimum frequencies.

Base station transmitter is amplitude-modulated and has an output of 200 w. It covers subscribers within a 200-mile range. Subscriber sets are operated on 12 v batteries and transmission is also amplitude-modulated.

Plastic Low Tension Insulator

As a substitute for porcelain insulators (formerly imported from Japan), Novex Pty. Ltd., Hobart, Tasmania, has developed a plastic low tension insulator in collaboration with the Hydro-Electric Commission of the state.

Among advantages claimed for the plastic product in comparison with the ceramic type are: Lower weight, higher strength, lower moisture deposit at high humidities, better finish and uniformity of the thread for attaching to the pin, and elimination of cracking caused by unequal expansion between pin and insulator body.

Puncture tests carried out by the Hydro-Electric Commission gave readings of 44 kv dry and 38.3 kv after hot and cold water tests. The insulator withstood a mechanical load test of 1,200 lb. These insulators are used on 415 v lines.

Air Navigation Instruments Checked Rapidly

More than 70 percent of time usually required for checking and regulating airplane navigation instruments is saved by a new apparatus used in Mexico City.

With high accuracy, the device checks gasoline, oil, anti-freeze and hydraulic reserve indicators, and, in general, all self-synchronized instruments. More than 1,000 ft of cable were used in construction of this tester and it is easily operated.

Four Stumbling Blocks to Economic Atomic Power

Nuclear energy will not contribute any large amount of power during the next ten years, according to Sir John Cockcroft, director of Britain's Atomic Energy Research Establishment. Sir John bases his opinion on estimates of well-informed scientists (see page 163, this issue) and four major technical problems still to be solved.

While natural uranium piles can generate power as a byproduct, their fuel efficiencies would be so low that they would not be economical. Primary fuel source, uranium 235, must be made to "breed" secondary fuel from uranium 238 or thorium. Materials that will resist higher temperatures without deforming must be developed. Efficient ways must be found to process the nuclear fuel and to remove the radioactive ash from the pile safely. The last major problem confronting nuclear power development is the final disposition of large quantities of radioactive ash.

Level Gage Measures Pulp Stock

A new type level gage has been developed in Australia to measure the height of pulp stock in blow tanks. Known types of gages, such as air bubble and differential gages, proved unsuitable for this operational purpose because the pipeline that connects the blow tank with the level gage becomes blocked by pulp.

The new instrument, which operates electrically, consists of a pressure head mounted in the blow tank wall at the point from which stock level is to be referred. The head is a stainless steel diaphragm that contacts the stock surface. On the upper side it carries an electrical device that translates pressure variations caused by the rise and fall of stock into electrical variations.

This plant also makes interesting use of quantitative measuring electrodes to estimate the quantity of soda ash passing down the drain.

Strain Tester for Rubber

A NEW DEVICE developed at the National Bureau of Standards conveniently measures strain in vulcanized rubber, in contrast to conventional tensile testers that determine tensile stress. In this apparatus strain is measured at a definite time after application of a predetermined stress, while usual testing methods use essentially the opposite procedure. Bureau investigations revealed that the new technique combines a higher degree of reproductibility in measurement with an accuracy not previously possible. These investigations were part of a program conducted for the Office of Rubber Reserve on the improvement of physical testing methods for synthetic rubbers.

During the past three years both strain and tensile tests have been made at the Bureau on several hundred sheets of various lots of GR-S, GR-I, and GR-M synthetic rubbers. Statistical analyses of these data have shown that variances introduced by the strain test are negligible compared to the variances in different sheets of rubber from the same batch, or in batches from the same lot of polymer. Differences introduced by the usual measurements of stress at a specified elongation are of the same order of magnitude as those for different sheets and different batches.

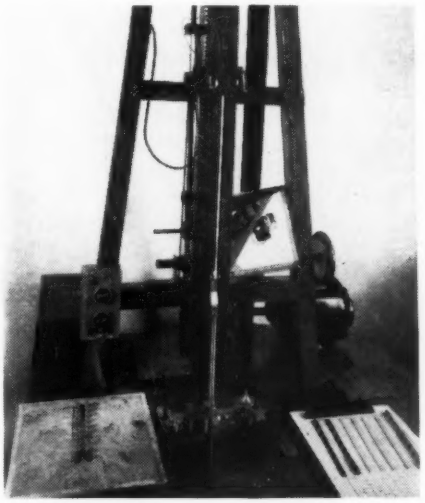
Since 20 specimens for the strain test can be cut from a single test sheet, it has now become possible to detect differences in the properties of rubber from different parts of the sheet, or to study the effect of aging on stiffness or modulus by means of a single test sheet. The relatively large distance between bench marks on the specimens used in the new test and the fact that the test subjects them to a uniform stress for a definite length of time make these specimens desirable for measurements of set. Values determined in this manner are found to be much more reproducible than the usual measurements made after rupture of the specimen.

The strain tester was designed to incorporate this improved reproducibility of measurements and high degree of accuracy in an apparatus suitable for routine testing. Design features responsible for the precision and accuracy are: (1) Observation of bench marks when they are essentially at rest; (2) use of freely suspended weights to attain the desired stress without frictional effects; and (3) increase in the distance between bench marks of four times that used for the usual dumbbell-shaped specimens.

The tester is composed of two parts, built into a supporting table. One part, above the table, includes a mechanism for extending the specimen, an optical system for observing the upper bench mark on the specimen from a seated position, a mechanism for aligning a milimeter tape with the bench marks, a time cycle controller for regulating the sequence of operations and a keyboard for selecting the load to be applied to the specimen. Beneath the table is the mechanism for applying the selected load. This mechanism employs eight solenoids for adding eight different weights singly or in combinations to a weight assembly attached to the specimen.

Rubber specimens are strips 0.254 in. wide and 6 in. long, cut from the usual 6 in. square test sheet. The specimen die consists of six razor blade strips clamped between metal spacers. Five specimens are cut in a single operation. The bench marks, placed on the specimens in the usual manner, are spaced 10 cm apart. Using a special thickness gage, the average thickness of specimens between bench marks is obtained by a single measurement. The gage reading is entered on the keyboard of the strain tester.

After placing specimen in the grips, the starting switch is pressed, causing selected weights to be added to the weight assembly and starting the motor which pulls the specimen upward. When the specimen lifts the weight assembly from its support, a switch starts the time cycle controller. At subsequent predetermined intervals this controller (1) halts the upward travel of the specimen by stopping the motor when the weight assembly has been lifted sufficiently to prevent it from coming back to rest on its base as the specimen continues to stretch, (2) signals the operator and lights the lamps in the optical system for observing upper bench mark and hair line, (3) extinguishes lamps to prevent late observations, (4) returns specimen and weights to their starting positions by starting the motor in reverse direction, and (5) stops itself at the end of the cycle.



RUBBER STRAIN TESTER from perspective of operator, shows the optical prism, rubber specimen in test position and the mirror in which upper bench mark is seen. Weight-selector keyboard is at lower left.

The strain test is not intended to measure stress-strain properties near or at rupture. For control testing, however, and for much research, a knowledge of the strain for any stress below the region of rupture is sufficient. The improved precision obtained in the strain test warrants a separate determination of the stress and strain at failure in those cases where they are necessary.

Secret British Casting Process Licensed to American Foundry

NEWEST TECHNIQUES of permanent mold centrifugal casting have been made available to American industry through license contracts recently signed between Firth-Vickers Stainless Steels, Ltd. and Lebanon Steel Foundry. The process has been termed the most important improvement in the casting of alloy steel during the past generation of foundry practice. It will be used immediately in the production of vital turbojet components needed in the 500,000,000 dollar U. S. military aircraft engine program.

Aside from its immediate importance to warplane engine production, it is said the process can be used in engineering applications in which highly exacting requirements of heat resistance and corrosion resistance must be met. Because of the unusual design characteristics and integrity of castings made by the process, the way is opened to production of cylindrical and tubular structures that are difficult or impossible to forge or cast by any other method, and that cannot be hollow-rolled.

The advent of turbojet gas turbine engines for aircraft purposes created wide and urgent need for circular engine components—rings, valves and sleeves inclosing the rotating turbine—to operate at greatly elevated temperatures. Castings made by this process have performed in extended jet

engine service without a single failure.

While maintaining secrecy of the casting process details, Lebanon officials have said that centrifugal permanent mold alloy casting for heat and corrosion resistant service is the casting of molten metals and alloys in heavily constructed molds while the molds are rotated about their axes at high speed. Molten metal is deposited and solidified under pressure within the mold, thereby producing a symmetrical cylindrical ring of high density with excellent structural properties throughout.

The process is specifically adapted to production of high-melting-point alloys and steels ranging from carbon steel grades to the new super alloy types. Some of these compositions are extremely heat-resistant and strong at high temperatures, but because of this heat-strength characteristic, they are difficult or impossible to forge into conventional shape.

Outstanding Men of Steel To be Picked by Industry

NOMINATIONS are now in order for living individuals who have made outstanding contributions to progress of alloy steels. The men selected will have their efforts recognized by Distinguished Service Awards to be presented during the "Salute to Alloy Steel" celebration at the 30th Annual Metal Congress and Exposition. Reason for these awards at this time, is that first engineering use of alloy steel occurred 75 years ago.

Emphasizing that men to be honored must be dealing with engineering alloy steels in contradistinction to tool and high alloy steels, the Awards Committee will consider nominations of outstanding men in all branches of this field, including: Those who have perfected metallurgical processes and equipment; discoverers of advanced research techniques; discoverers of improved inspection techniques; discoverers and perfectors of the alloy steels; notable men in the organization and promotion of the alloy steel business, and men responsible for notable applications in consuming industries.

Reader's nominations should be sent to J. M. Schlendorf, Chairman, Distinguished Service Awards Committee, c/o Republic Steel Corp., Republic Building, Cleveland 15, Ohio.

TURBINE WHEEL with 3 deg conical form and rim loading bursting at 52,000 rpm. Wheel made of 14ST aluminum.

SIMILAR TURBINE WHEEL but with hubs on both sides and no rim loading burst at 62,230 rpm in spin pit.

Destructive Testing Proves Jet Engine Components

TWO "SPIN PITS", which whirl turbine components for new jet engines at speeds up to 100,000 rpm at temperatures of 1,750 F, have recently been disclosed by the Boeing Airplane Co. propulsion laboratory. They are used for proving parts for jet engines now under development.

The pits are made of layers of 4½ in. steel plate with 3½ in. maple blocks mounted on the inside. Specimens as large as 20 in. in diameter are driven by a small compressed air turbine. A hoist, supported on a steel pylon behind each pit, supports the heavy weight of the cap plate, air turbine and accessories. Once the test material has been put in position, the assembly is lowered into the pit, clamped in place and the air in the pit is evacuated.

One pit has built-in heating coils of nichrome steel wire embedded in clay holders to maintain constant temperatures up to 1,750 F. Automatic controls keep temperature to pre-set levels and automatic recorders retain all test measurements. Weldment of turbine blades, design of hubs, types of metal and design configurations are accurately tested under operating conditions as separate units.

High speed photographs, made with an ordinary 8 x 10 view camera, enable engineers to study test wheels as they break. A small mercury lamp is the light source for the one five-millionth of a second flash. The light is tripped by double, concentric copper screens that line the pit. At burst, pieces of the test wheel crash into the screens, push them into contact and complete the flash circuit. With this technique, any portion of the wheel that breaks loose takes its own picture and establishes the point of failure.

An electronic revolution counter measures the speed of the wheel. One end of the spinning turbine shaft is divided into two sections: Half black, half white. A photoelectric cell, focused on the shaft, sends a current impulse to recording instruments each time the black portion passes its scanner. Timed and checked electronically, the speed is controlled to within one-half of one percent right up to the time of fracture.

Spin pit results are showing the Boeing engineers effects produced on turbine wheels by: Weight and tensile strength of materials, rim loading of blading, cross-section shape and design of shaft holes in hubs.

Free Course Aims at Improving Specification of Springs

DESIGN ENGINEERS who have occasion to specify springs are invited to attend a new two-day refresher course on spring design, tolerances and specifications being held during alternate weeks at the plant of the Hunter Spring Co., Lansdale, Pa. The first course was given on August 31, 1948.

The free course is being offered because of the conviction that spring specification today is largely done without sufficient attention to the service that can be expected of a spring and the dollar economies possible by more careful attention to design details. Company officials further feel that designers are largely unaware of the tolerances and precision that can and cannot be attained.

The course includes demonstrations of tolerance computation, open forums and discussions, plant visitations, and six lectures on: Basic spring design, the checking of spring specifications, controlling the test load, spring materials and stresses, do's and don'ts in tolerance specification, and inspection and testing methods.

Here is the agenda of the course:

Tuesday am
- 8:30 — 9:00, preliminaries
- 9:00 — 12:00, plant tour

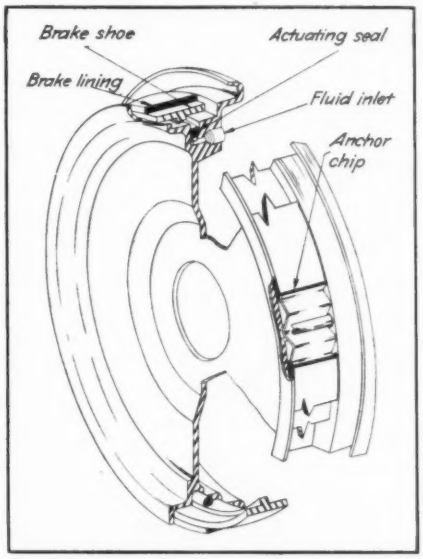
Tuesday pm
- 1:15 — 3:45, "Basic Spring Design"
- 4:00 — 5:00, "Checking Spring Specifications", demonstrations
- 7:00 — 8:00, "Controlling the Test Load"
- 8:15 — 9:00, "Computing Tolerances", demonstrations

Wednesday am
- 8:30 — 9:45, "Spring Materials and Stresses for Design"
- 10:30 — 11:45, "Inspection and Testing"

Wednesday pm
- 1:15 — 3:30, "The Drawing Specifications and Tolerances"
- 3:45 — 4:30, Forum and Discussion.

There are no fees or charges for the Hunter design course. The only expenses involved are those for traveling to and from Lansdale and for living while at Lansdale. Room reservations at the Hotel Tremont in Lansdale will be handled by Hunter on request.

Engineering executives who feel that their designers should attend, or who are asked by designers to make arrangements, should apply to Hunter Spring Co. (R. P. Coleman, Jr.) so that convenient dates can be set.



TEST AUTOMOBILE BRAKE of new design shows anchor clips between working surface segments of the brake lining. Five working segments provide large braking surface and reduce lining wear.

New Automobile Brake Adapted from Airplane Design

A RADICALLY NEW AUTOMOBILE BRAKE, with 50 percent fewer parts than in the conventional type; faster, more positive action; greater braking surface and other advantages was announced recently by the Glenn L. Martin Co. The new brake is an adaptation of a new type of hydraulic brake the company developed during the war for its airplanes. Tests of the new brake under actual road conditions have been highly satisfactory.

No wheel cylinders, pistons and linkages are needed in the new brake. Consequently it should be suited for automobile manufacturers following the trend toward the use of smaller wheels with less room for brake mechanism.

The brake uses a continuous ring seal that fits in a groove in the brake shoe support. Hydraulic fluid, actuated by the brake pedal, enters this groove under the seal. The seal is forced outward and actuates the brake shoe in virtually a 360 deg engagement with the drum. Ring-shaped arrangement of the power applying seal provides for simultaneous application of equal braking force around the brake drum surface. This gives better wear characteristics and adds to the lining's life expectancy.

Full movement of the actuating seal of the brake is attained with relatively little fluid. This allows for extreme sensitivity or responsiveness to pressures caused by operating the pedal.

MEETINGS

October 5-7
NEW ENGLAND MATERIALS EXPOSITION—Mechanics Hall, Boston.

October 11-13
AMERICAN SOCIETY OF TOOL ENGINEERS — Semi-Annual Convention, Biltmore Hotel, Los Angeles, Cal.

October 12-13
PACKAGING MACHINERY MANUFACTURERS INSTITUTE—Annual Meeting, Hotel Roosevelt, New York, N. Y.

October 12-16
AMERICAN CHEMICAL SOCIETY—National Chemical Exposition and National Industrial Chemical Conference, The Coliseum, Chicago, Ill.

October 13-15
MAGNESIUM ASSOCIATION—Mid-Year Meeting, Skytop Club, Skytop, Pa.

October 13-15
PORCELAIN ENAMEL INSTITUTE—Annual Forum, University of Illinois, Urbana, Ill.

October 14-15
GRAY IRON FOUNDERS' SOCIETY—Annual Meeting, Haddon Hall, Atlantic City, N. J.

October 18-22
AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS—Midwest General Meeting and Technical Session on Tracer and Contouring Control System for Machine Tools. Hotel Schroeder, Milwaukee, Wis.

October 21-23
OPTICAL SOCIETY OF AMERICA—Annual Meeting, Hotel Fort Shelby, Detroit, Mich.

October 22-25
METAL TREATING INSTITUTE—Annual Meeting, Adelphia Hotel, Phila.

October 25-29
NATIONAL METAL EXPOSITION—Commercial Museum and Convention Halls, Philadelphia, Pa.

AMERICAN SOCIETY FOR METALS—Annual Convention, Benjamin Franklin Hotel, Philadelphia, Pa.

AMERICAN WELDING SOCIETY—Annual Convention, Bellevue-Stratford Hotel, Philadelphia, Pa.

AMERICAN INSTITUTE OF MINING AND METALLURGICAL ENGINEERS—Annual Fall Meeting of Inst. of Metals Div., Hotel Adelphia, Phila., Pa.

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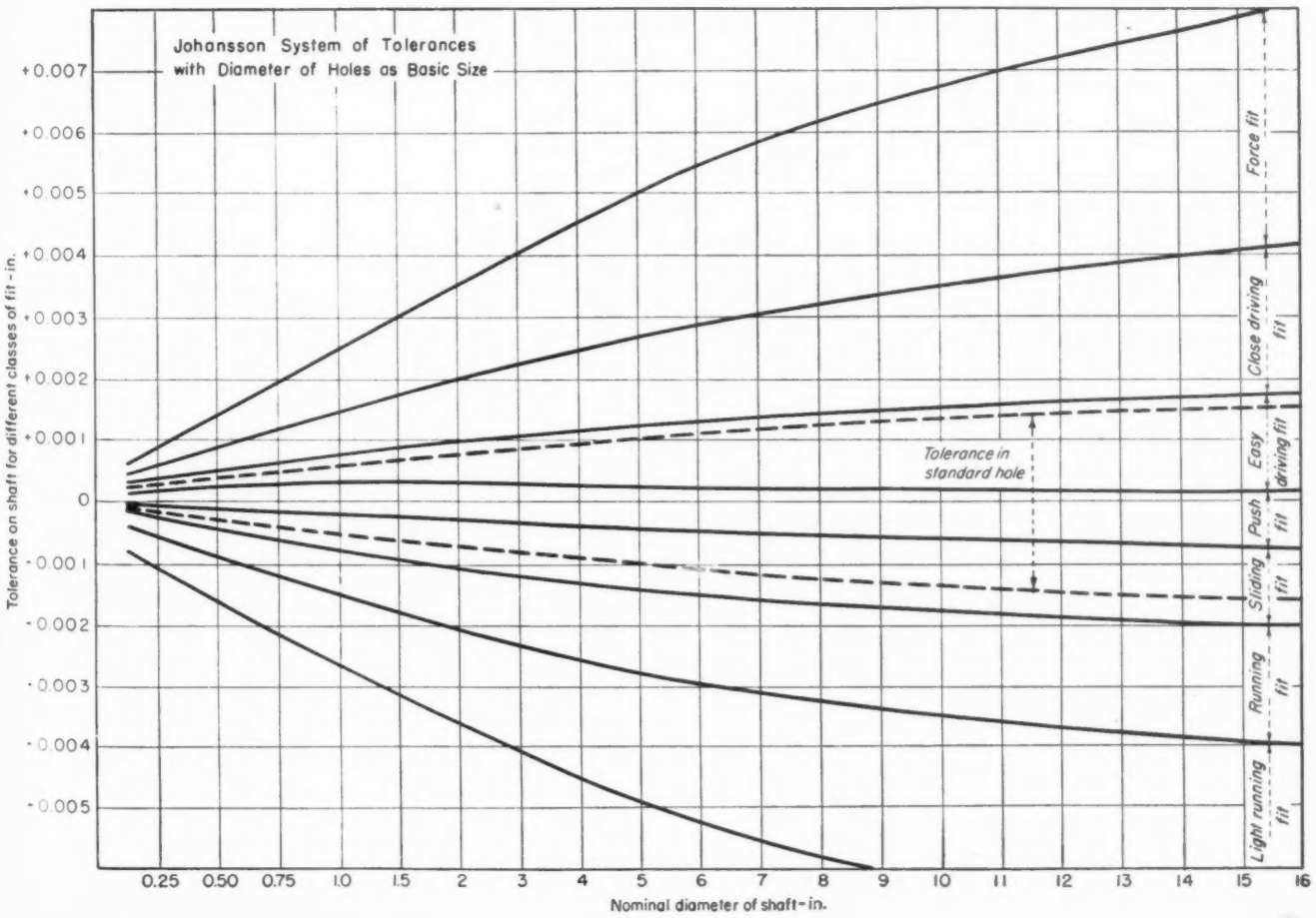
Graphical Representation of the Johansson System of Machining Tolerances

HAROLD WOODHOUSE
Executive Engineer, General Engineering & Research Corporation

THE JOHANSSON SYSTEM of machining tolerances for the various classes of fits between parts to be assembled is often used in this country for mass produced parts and for custom or medium production items. It is based on the assumption that the hole is always made the nominal size with a fixed tolerance and that all of the machining tolerance deviation from the nominal size is made on the shaft or male part to produce the desired fit.

These tolerances are listed in many handbooks in tabular form and usually only one tolerance is given to cover a range of several inches in the larger sizes. For this reason, reducing the tolerances to chart form makes instant visual reference possible and shows clearly how the class of fit varies with tolerance for a given size.

Chart covers all ordinary types of fits. If extremely close adherence to the quality of fit is required, the tolerance shown for the standard hole on the chart by the ordinate between the two dotted lines is reduced by 50 percent. Note that the hole tolerance is equal on both the plus and minus side of the nominal.



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RADIO WIRE

- Lead-in Wire
- Hook-up Wire
- Wiring Assemblies
- Aerial Wire
  Bare, Tinned or Enameled
- Bonding Straps
- Plastic Insulated

FLEXIBLE CORD

- All Synthetic Rubber Twin Cord
- Cotton Braided Cord
- Synthetic Rubber-Sheathed Cords
- Shielded Synthetic Rubber-Sheathed Cords

AIRCRAFT CABLE

STARTER CABLE

- To Army-Navy Specification AN-C-161 aluminum conductor
- To Army-Navy Specification AN-J-C-48-A copper conductor

HIGH TENSION IGNITION CABLE

- Steelductor to Army-Navy Specification AN-C-130A, 5mm, 7mm and 9mm
- Copper conductor to Specifications AMS-3390, AMS-3392 and commercial standards

LIGHTING CABLE—all gauges

- To Army-Navy Specification AN-C-161 aluminum conductor
- To Army-Navy Specification AN-J-C-48-A copper conductor
- Shielded to Specification ANC-168
- Wiring Assemblies

AUTOMOTIVE TYPE CABLE

HIGH TENSION IGNITION CABLE

- Steelductor
- Copper Conductor
- Shielded

BATTERY CABLE—all gauges

- Rubber Covered
- Lacquered Insulated
- Tinned Braided Strap
- Tinned Rope

LIGHTING CABLE

- Synthetic Rubber and Plastic Braided and Lacquered
- Loom Braided and Lacquered
- Shielded

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H. R. ROSS
Research Division, United Shoe Machinery Corporation

THE CHARTS PRESENTED here are designed to minimize the mathematical calculation in analyzing the characteristics of either a harmonic or cycloidal cam curve of any proportion. The necessary information is reduced to total throw and elapsed time for that throw. The kinematics of motion at any instant are simple functions of these data.

For any given throw L, and time T, corresponding to a complete harmonic or cycloidal cam motion, the displacement at any time from start can be read directly from the curves as a percentage of total displacement, maximum velocity and maximum acceleration.

Maximum throw will always be known and maximum velocity and acceleration for harmonic and cycloidal cams are determined from the following formulas:

Harmonic Cams—

$$\text{Max velocity} = \frac{\pi L}{2T} \text{ in. per sec} \quad (1)$$

$$\text{Max acceleration} = \frac{\pi^2 L}{2T^2} \frac{\text{in. per sec}}{\text{per sec}} \quad (2)$$

Cycloidal Cams—

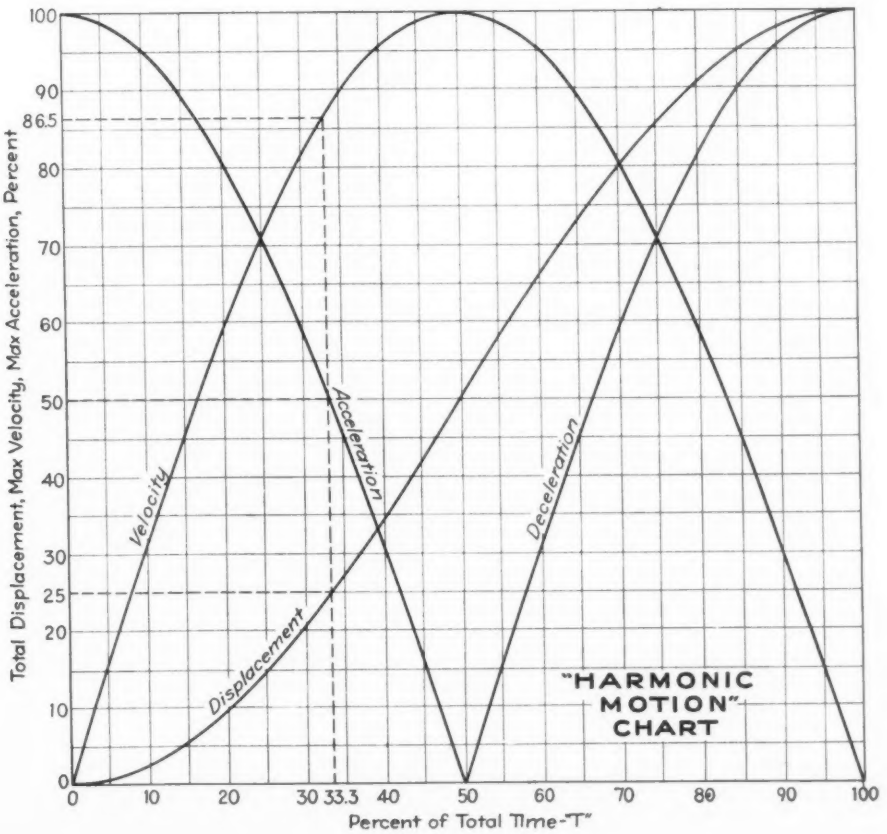
$$\text{Max velocity} = \frac{2L}{T} \text{ in. per sec} \quad (3)$$

$$\text{Max acceleration} = \frac{2\pi L}{T^2} \frac{\text{in. per sec}}{\text{per sec}} \quad (4)$$

Complete displacement-time, velocity-time and acceleration-time curves can be constructed for individual cams as percentages of total throw. Such curves are useful for making timing charts, clearance diagrams and velocity or acceleration curves, particularly as a method for comparing proposed motions or apportioning cycle time. They can also indicate where a smooth blend of different base curves can be made. The junction point of the two curves must naturally have equal values for displacement and velocity. A cycle can start with harmonic motion for fast initial acceleration and change to cycloidal motion for zero velocity and acceleration at the finish.

Following examples illustrate how to find one point on each of the percentage curves for both harmonic and cycloidal cams.

EXAMPLE 1. A harmonic cam with 1 in. throw in 90 deg is rotating at 100 rpm. Find displacement, velocity and

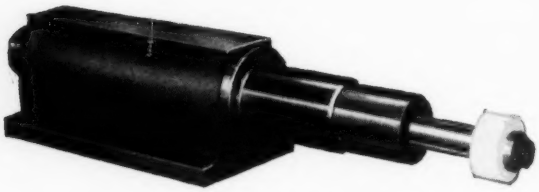


POPE

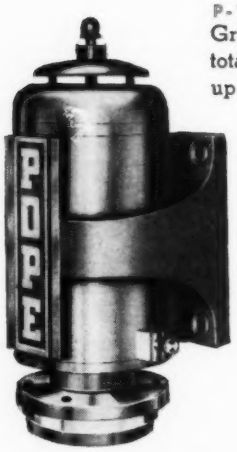
GRINDER SPINDLES AT A GLANCE

P-327 Pope Sealed Package Spindle for 6" x 18" Surface Grinders, with a full 1 HP, 3450 RPM totally enclosed motor; for use with grinding wheels up to 8" O. D., ¾" face, 1¼" hole.

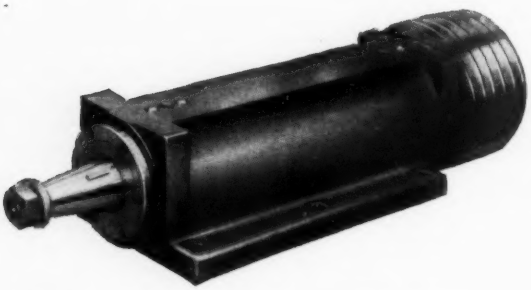
P-1004 Pope Direct Motorized Tool Grinder Spindle with 1 HP, 3450 RPM totally enclosed motor; for grinding wheels up to 8" O. D., ¾" face, 1¼" hole.



P-2463 Pope Heavy Duty Vee Belt Driven Cartridge Type Spindle in mounting bracket; for grinding wheels up to 4" O. D. at 6000 RPM, 2" face with 1" hole.



P-2500 Pope Heavy Duty Motorized Spindles in a variety of sizes from ¾ HP to 20 HP and from 900 to 3600 RPM.



P-666 Pope Heavy Duty Vee Belt Driven Grinding Machine Wheel Head Spindle; for grinding wheels up to 28" O. D., 1½" face, 8" hole at 750 RPM. Maximum speed 4000 RPM.

Here is a representative showing of POPE Precision Grinding Spindles. Others are available in a great variety of sizes and styles. Each assures:

1. Constant Production of Uniformly Good Finished Surfaces.
2. Cool Operation At High Speeds Over Long Periods.
3. Sealed Lubrication Performance — no oiling or greasing, nothing to renew, replace or adjust throughout the entire life of the bearings.

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acceleration when cam has rotated 30 deg from the start. (It is assumed that the output is pure harmonic and that for a swinging follower or offset roller, the came has been corrected for arc swing to make machine time agree with cam time.)

SOLUTION. With total throw L equal to 1 in., time T elapsed in rotating 90 deg is

$$T = \frac{90 \times 60}{100 \times 360} = 0.15 \text{ sec}$$

At instant in the cycle 30 deg from the start, point at 33.3 percent of total time on Harmonic Motion Chart is considered. From displacement curve on chart, percent of total throw is 25 percent, or 0.25 in.

Max velocity from Eq (1) is

$$\frac{\pi L}{2T} = \frac{\pi \times 1}{2 \times 0.15} = 10.5 \text{ in. per sec}$$

Considering the same point for percent of total time, the velocity from chart is 86.5 percent of maximum or, 9.1 in. per sec.

Max acceleration from Eq (2) is

$$\frac{\pi^2 L}{2T^2} = \frac{\pi^2 \times 1}{2\,(0.15)^2} = 200 \text{ in. per sec per sec}$$

Similarly, the acceleration from the chart is 50 percent of maximum, or 110 in. per sec per second.

EXAMPLE 2. Find similar values for a cycloidal cam having same constants, but at a point 50 deg from start.

SOLUTION. As in Example 1, L equals 1 in. and T equals 0.15 second. Referring to the Cycloidal Motion Chart and considering the point 50 deg from the start or at 55.5 percent of total time, T, displacement from chart is 61 percent of maximum, or 0.61 in.

Max velocity from Eq (3) is

$$\frac{2L}{T} = \frac{2 \times 1}{0.15} = 13.3 \text{ in. per sec}$$

Actual velocity from chart is 97 percent of maximum, or 12.9 in. per sec.

Maximum deceleration from Eq (4) is

$$\frac{2\pi L}{T^2} = \frac{2\pi \times 1}{(0.15)^2} = 279 \text{ in. per sec per sec}$$

Actual deceleration from chart is 34 percent of maximum, or 95 in. per sec per second.

EXAMPLE 3. To accelerate with a harmonic curve as in Example 1 for 45 deg and decelerate with a cycloidal curve during the next 45 deg, both curves to blend at maximum velocity.

Where L_1 is the throw for a complete cycloidal curve that will blend at its mid-point to the harmonic curve of Example 1, and since time T is specified to be the same for each curve, Eq (1) and Eq (3) are equal.

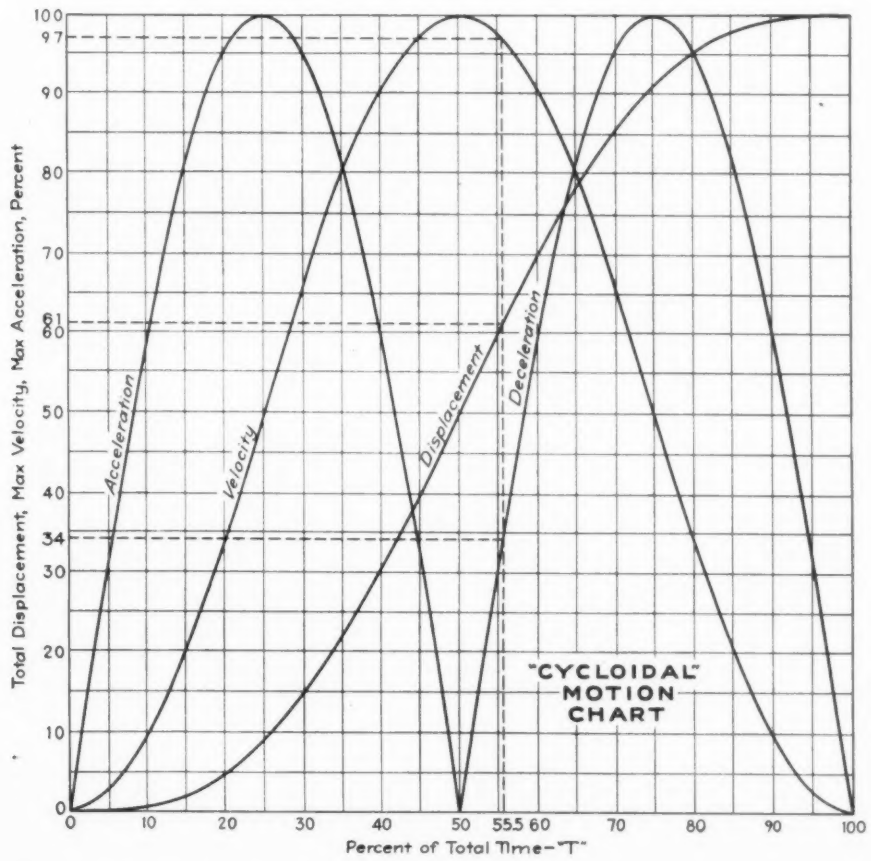
$$\frac{\pi L}{2T} = \frac{2L_1}{T}$$

$$L_1 = \frac{\pi L}{4} = 0.786 \text{ in.}$$

Since L is given as 1 in., total throw of the combination is

$$\frac{L + L_1}{2} = \frac{1 + 0.786}{2} = 0.893 \text{ in.}$$

With the values for T, L and L_1 known, a complete analysis of the combined curve is readily available from the charts.



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NEW MATERIALS AND PARTS

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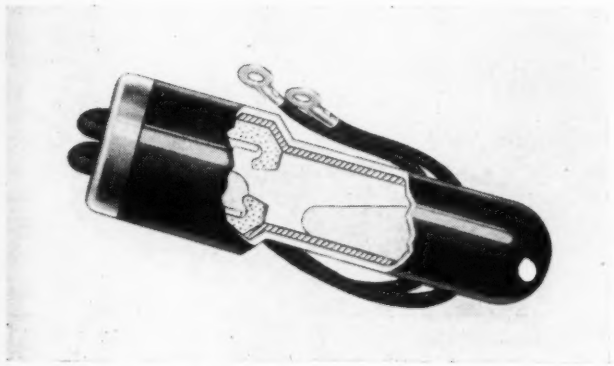
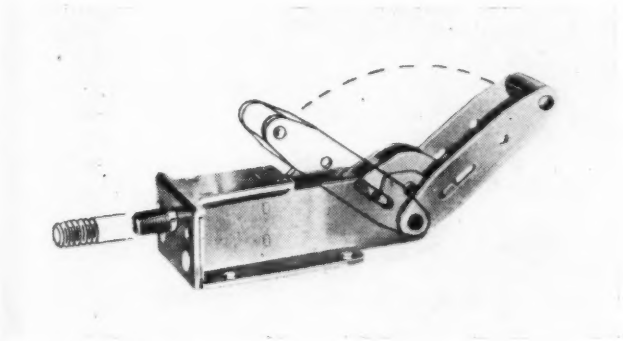


Metal-Cased Mercury Switches

Durakool Inc., Elkhart, Ind.

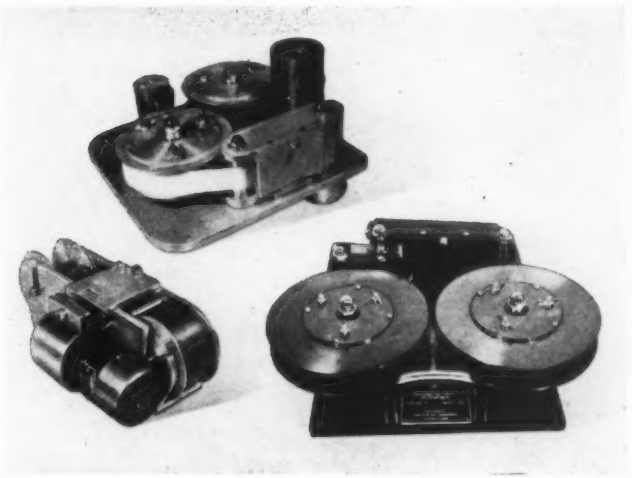
Electrode corrosion is prevented by hydrogen gas sealed by an electric weld in the metal case of these mercury switches. Standard models have one electrode, the case being the second, while the Double-Flow model make and break contacts are between two mercury pools. Units are applicable in high capacity, highly inductive circuits with loads ranging from 1 to 65 amp. An outer case of plastic or rubber can be supplied for making the switches both attractive and waterproof.



Push-Pull Clamp

Lapeer Mfg. Co., Lapeer, Mich.

Model P&P-800 clamp is designed to meet conditions not satisfied by conventional toggle clamps. It is intended for use in locations where space is limited or under circumstances where the operating handle cannot be accommodated except at some distance from the point of pressure application. Clamp can exert up to 800 lb pressure with normal hand operation in either the push or pull movement. Handle automatically locks and holds work securely upon completion of travel. Device can function as a clamp with the plunger at either extreme, or it can be used to exert a continuous pushing or pulling motion during the entire $1\frac{1}{4}$ in. travel of the plunger. The plunger can be supplied with both an external $\frac{3}{8}$-24 thread and an internal $\frac{1}{4}$-28 thread.



Magnetic Tape Recorders

Cook Research Laboratories, 1457 *Diversey Pky., Chicago* 14, *Ill.*

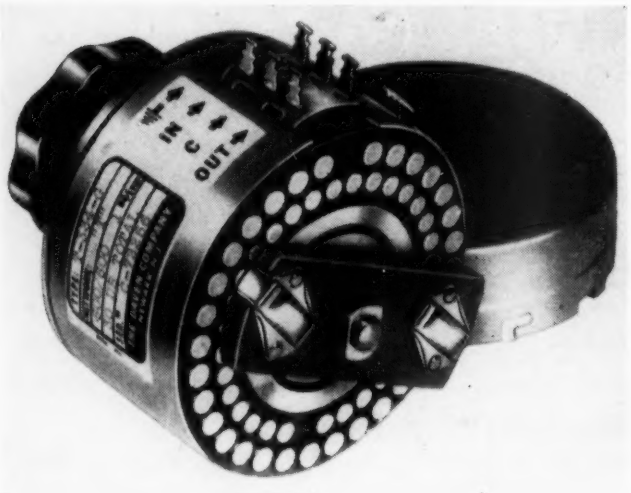
Three types of data recorder for application in vehicles, aircraft, guided missiles or other mobile equipment utilize magnetic tape to store variable or transient data under conditions of severe shock acceleration. All three have multi-information channels and include a time-base channel for speed and error compensation. Type MR-2 (top of illustration) is a 12-channel recorder having a recording time of 15 minutes. It is applicable to flight tests of engine temperatures, accelerations, strains, etc. Type MR-3 (lower left) is a 6-channel miniature instrument with a recording time of 3 minutes. Measuring $6\frac{1}{2}$ x $3\frac{1}{8}$ x $2\frac{1}{4}$ in. and weighing 24 oz, unit is suitable for

the measurement of test data in guided missiles. Type MR-6 (lower right) is a standard 6-channel recorder with a recording time of 25 minutes.

Interchangeable-Hub Sheaves

Transmission Machinery Corp., Box 7823, Dallas, Tex.

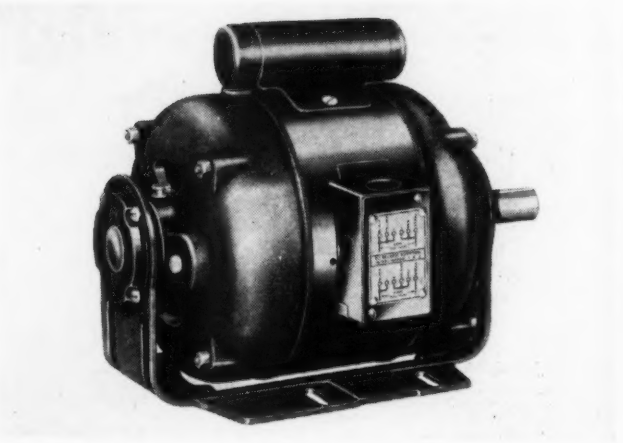
Sheave assemblies permit use of different diameter rims on the same hub and/or hubs of different bore on the same size rim. Three heat-treated socket-head screws and lockwashers secure the assemblies. Sheave rims are made of light-weight, semi-steel castings that are machined and balanced to insure maximum belt life. Applicable to drives of $\frac{1}{3}$ to 10 hp, units are available with one, two or three grooves for "A" and/or "B" section belts in sizes from 2.2 to 18.4 in. pitch diameters.



Compact Rotary Switches

The Daven Co., 191 Central Ave., Newark 4, N. J.

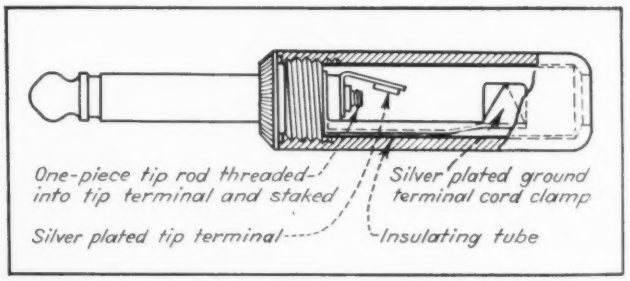
Relatively short, individually spring-loaded rotor blades permit the use of as many as six poles on one deck within a diameter of $2\frac{3}{4}$ inches. Uniform pressure on contacts and slip ring assure smooth wiping action and a low contact resistance of 0.004 ohm. Switches are rated at 15 amp, non-inductive, and are suited for use in bridge circuits, thermocouple systems, measuring instruments and strain gage apparatus. For use in power circuits, relay cutouts are required to prevent contact burning.



Rubber-Mounted Motors

Kingston-Conley Div., The Hoover Co., North Plainfield, N. J.

Fractional horsepower a-c motors mounted in resilient rubber provide quiet operation for heavy-duty applications. Units are rated up to $\frac{1}{2}$ hp at speeds from 1,725 to 3,450 rpm and are available with either single or polyphase windings. Both types can be furnished with either sleeve or ball bearings. Sleeve assemblies are packed with wool, while the ball bearings are prelubricated and sealed. A centrifugal "on-off" switch having oversized silver contacts can be supplied if desired.


One-piece tip rod threaded into tip terminal and staked
Silver plated ground terminal cord clamp
Silver plated tip terminal
Insulating tube

Redesigned Jack Plug

Switchcraft, Inc., 1328 N. Halstead St., Chicago 22, Ill.

Positive, quick connections can be made with this $\frac{1}{4}$ in. jack plug used in recorders, communications systems, testing equipment and related apparatus. The dual-purpose sleeve terminal clamps over the metal braid of shielded-wire cables and is easily soldered for a perfect electrical connection. One piece tip rod threads into the tip terminal and is staked to insure tightness. An interlock between terminal and body prevents shifting of the assembly. Plug is available with red or black tenite handles or with bright nickel-plated handles for shielding.

Non-Corrosive Thermoplastic

United States Rubber Co., Rockefeller Center, New York, N. Y.

Versalite is a combination of synthetic rubber of the acrylonitrile group with a synthetic resin. It is non-corrosive and stable up to 220 F, is tough and easy to form, and has good dielectric and low heat conductivity characteristics. Because of high impact and low water absorption properties, material is being used for a new

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PUMPS • FARM EQUIPMENT • MAGNETOS
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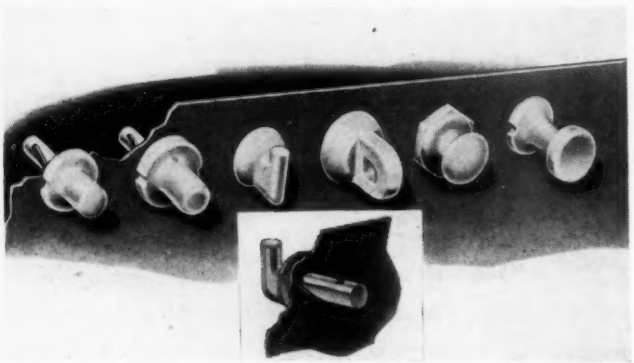
type shipping container for dry and frozen foods, as well as in home freezers, and related products. It can be made into irregular and compound shapes as large as five by ten feet in almost any thickness above 0.020 in.; can have solid color throughout with dull, satin, gloss or embossed finish. Plastic will not chip or warp and is resistant to gasoline, oils and most chemical cleaners.

General-Purpose Counters

Veedor Root, Inc., Hartford 2, Conn.

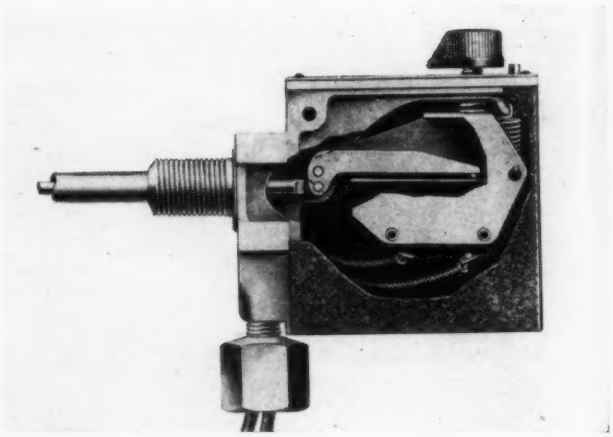
Four types of Series 1260 general-purpose counter cover the range from 1,000 to 5,000 counts per minute. Ratchet type permits 32 deg of throw to operate, while rotary ratchet type allows 45 deg of throw with 20 deg overtravel without registering more than one count. All types have standard reset and shaft can be at either end for clockwise or counterclockwise rotation. Overall dimensions are: $4\frac{1}{2}$ in. long, $1\frac{9}{16}$ in. high and $1\frac{1}{4}$ in. wide.



Plastic Fittings

Simmons Fastener Corp., Albany, N. Y.

Plastic Spring-Lock fittings lock and unlock with a quarter turn. Made of polystyrene, ethyl cellulose or butyrate and with steel inserts, devices can have a variety of head shapes. Applications include refrigeration shelf supports, dashboard plugs and ornamental knobs. The spring wire, which is twisted at installation, provides high tensional strength for withstanding vibration and permits rapid installation. Fittings can be inserted from one side of panel so that both original installations and replacements can be made in "blind" fashion.



Watertight Thermostat

Alloy Bellows Engineering Co., Cleveland 12, Ohio

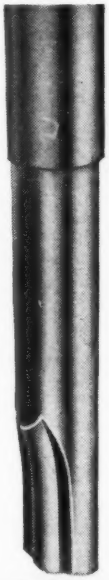
Bordee Type K-1 close differential-thermostat has been developed for use in processing equipment where excessive moisture or dust are present. Metal case is securely soldered to cast brass body and lead wires are sealed with special cement. Temperatures between sub-zero and 300 F can be withstood. Four different lengths of thermal element ranging from 6 to 24 in. have differentials between ± 2 and $\pm\frac{1}{3}$ F and adjustable ranges between 90 and 20 Fahrenheit. Electrical contacts are snap acting with a variety of arrangements and have a capacity of 10 amp, 125 v or 5 amp, 250 v, alternating current. Outlet is standard $\frac{1}{2}$ in. pipe thread suitable for use with flexible armored cable or conduit.

Bridge Amplifier

Ruge-de Forest, Dept. Pre. E, 76 Massachusetts Ave., Cambridge 39, Mass.

A sensitive deflection-type amplifier is designed for use with resistance bridges ranging in impedance from 100 to 500 ohms. Instrument has a full-scale output as small as 25 microvolts per volt of input to the bridge, or as large as 5 millivolts per volt. It is therefore suited for use with a wide variety of commercially available pickups for the measurement of fluid pressure, load, torque, stress or strain, temperature, and other variables. Output is 18 v across an external 1,500 ohm load with a linearity of better than $\frac{1}{2}$ percent. Maximum sensitivity is such that, with a 120 ohm resistance wire strain gage bridge, full output is obtained for a strain of 50 microinches per inch applied to only one arm of the bridge. A 1,200-cycle

TYPE "A" ASSEMBLY BIT



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Mr. President: Increased production is the key to lower costs. When presidents of a wide variety of other concerns say they have scored assembly production increases ranging from 15% to 50% . . . and when they credit these increases to change-over to CLUTCH HEAD . . . this modern screw should merit your investigation.

Mr. Engineer: Your selection of fasteners is important to the speed, safety and smooth operation of assembly . . . also to simplified field service, so vital to the "job performance" of your product. CLUTCH HEAD invites you to make a comparison on these points *with all other* screws on the market . . . recessed heads as well as slotted.

Mr. Purchasing Agent: Your experience tells you that the cost of screw APPLICATION is the factor that determines lower or higher FINAL COST in assembly. Users attest that CLUTCH HEAD'S exclusive features reduce costs with a 3-way step-up in production . . . faster, safer and easier driving; plus near-zero in tool maintenance cost.

Mr. Superintendent: Your main concern, and that of your foremen, is a smooth, safe, and steady tempo on the line. The high visibility of CLUTCH HEAD's recess is a speed factor, even with "green" operators; dead-center entry checks out driver canting and chewed-up heads; the non-tapered driving engagement (free from "ride-out") eliminates risk of skid damage and disposes of operator fatigue from fighting end pressure.

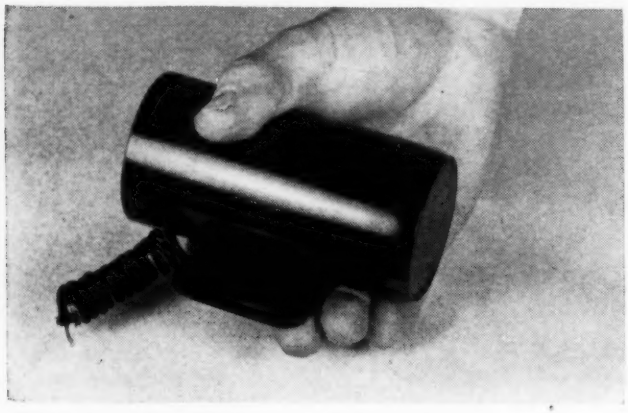
Gentlemen: These and *other exclusive advantages of CLUTCH HEAD will be apparent to you on examination. May we mail you screw assortment, sample Type "A" Bit, and illustrated Brochure? Check all the above features! also the ruggedness of the Type "A" Bit for longer continuous service and simple 60-second reconditioning; the value of the Lock-On for one-handed reaching; and CLUTCH HEAD'S basic recess design for common screwdriver operation.*

CLUTCH HEAD · SCREWS ·

UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 CHICAGO 8 NEW YORK 7

source of bridge power, complete bridge balance controls and balance indicator, precision attenuator, internal calibration, and precision phase-sensitive rectifier are features of the instrument.



Motor Capacitors

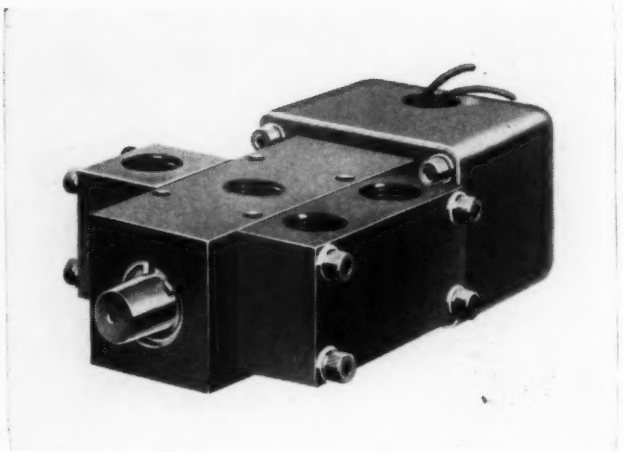
General Electric Co., Schenectady 5, N. Y.

Pyranol motor capacitor enclosed in 10-gage steel tubing is designed to withstand the same physical treatment as the motor on which it is operated. Bushings are made of silicone material that seals permanently by compression without gaskets or stickers. The long Flamenol insulated leads, unaffected by baking heat, oil, water, or mild acids, are protected by insulating eyelets held in the enclosure openings. Unit is available in all standard ratings.

Split-Armature Relay

Kurman Electric Co., 35-18 37th St., Long Island City 1, N. Y.

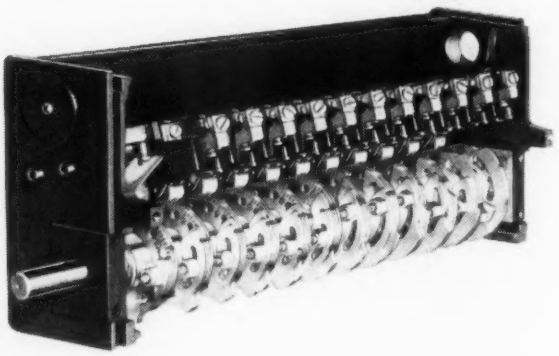
Type 24 relays embody a magnetic circuit of high permeability nickel alloy. The resultant steep magnetic curve insures dependable, constant operation at low current valves. Split-type armature permits simplicity and compactness of design. Rated sensitivity is 0.014 watt a-c, or 0.3 v-a a-c; with adjustability at 0.005 watt from 0.01 to 115 v, direct or alternating current. Flexibility of contact combinations makes this relay adaptable to a variety of uses for automatic controls, keying, antenna changeover, burglar alarms and closed circuit applications.



Four-Way Selector Valves

Saval Inc., 1915 E. 51st St., Los Angeles 11, Calif.

Series 8900 a-c solenoid-operated 4-way selector valves can handle water, oil or air at operating pressure up to 250 psi. Of two-position, spring-loaded type, valves have special metal-to-metal sealing members that wipe away foreign material. Conversion to d-c operation can be made by changing solenoids. Assemblies measure $7\frac{13}{16}$ x $5\frac{1}{4}$ x $3\frac{1}{2}$ in. and ports can be tapped to fit $\frac{1}{4}$, $\frac{3}{8}$, $\frac{1}{2}$, $\frac{3}{4}$ and 1 in. standard pipe threads.



Rotating-Cam Limit Switch

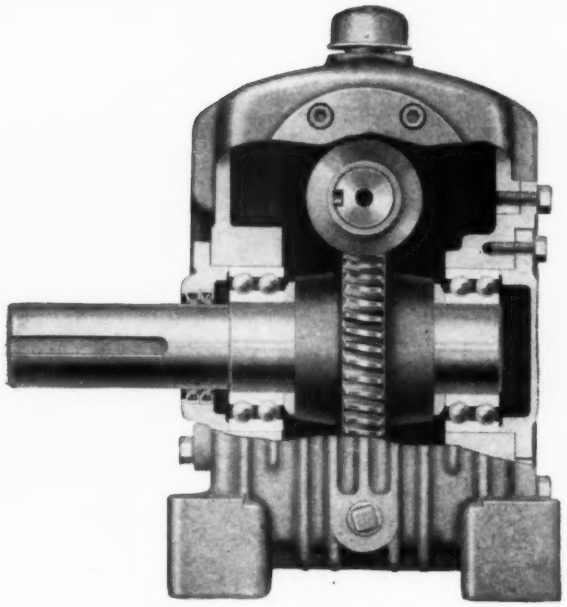
Allen Bradley Co., Milwaukee, Wis.

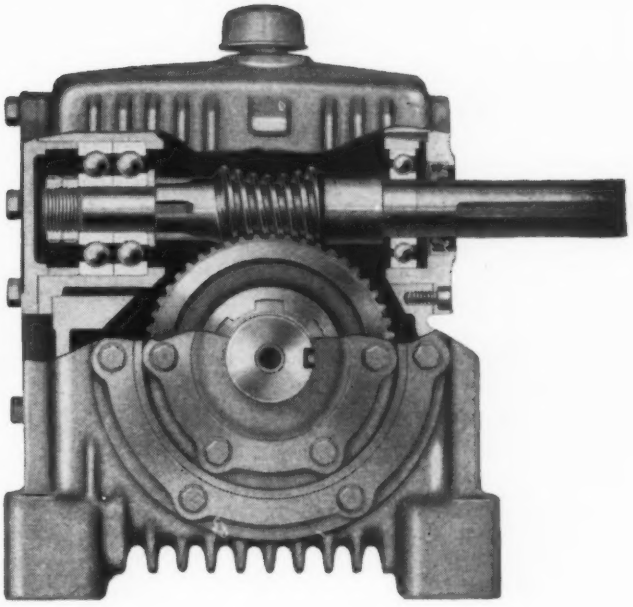
Bulletin 801 limit switch opens and closes up to 12 circuits at any position of the actuating cam shaft. Switches are also available for 3, 4, 6, or 9 circuits. All circuits can open or close in any sequence or elapsed time. Each cam is indexed for accurate setting. Contactors are individually detachable, and shaft and cam assembly is easily removed for adjustment. Switch is adaptable to any programming control or time-sequence operation. Available in general-purpose or watertight enclosures.

Corrision Resistant Wood Finish

Carboline Co., 502 N. Taylor, St. Louis 8, Mo.

Wood, rubber, carbon and fiberglass surfaces coated with Carbo-Kote are resistant to most acids, alkalis, solvents and mixtures of such agents. Chief exceptions are nitric acid and sulfuric acid of more than 45 percent concentra-





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3 HP with a 2" GEAR*SET?

632 HP with an 18" GEAR*SET?

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Impossible? These are the actual ratings of some Cone-Drive gear sets at 1750 rpm. The larger gear set can actually handle 600,000 INCH POUNDS of torque at 100 rpm.

These *standard* Cone-Drive gear sets and speed reducers—with their greater contact per tooth and MORE TEETH IN CONTACT—are available in ratios of 5:1 to 70:1 and from fractional hp to hundreds of hp.

Write for complete information today, giving your approximate hp or torque and speed requirements.

CONE-DRIVE

CONE-DRIVE DIVISION MICHIGAN TOOL COMPANY
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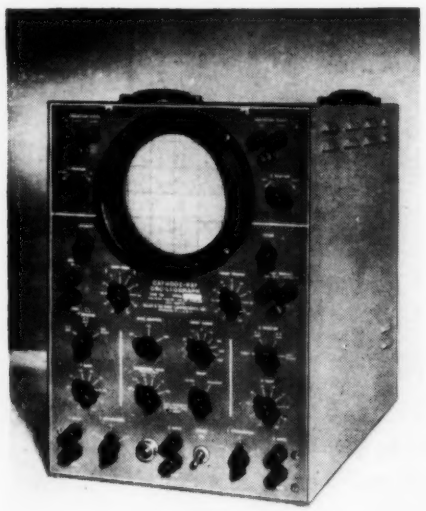
tion. Finish is smooth, wear resistant and waterproof, and is unaffected by temperatures up to 325 F. After application with a brush at room temperatures, the viscous preparation sets by polymerization, leaving a film that will flex about 10 to 15 percent. This finish has been used on wood tanks, floors, racks, tables and fume hoods.



Three-Channel Oscillograph

Allen B. Du Mont Laboratories, Inc., Clifton, N. J.

Type 250 cathode-ray oscillograph has three different channels through which signals can be applied to the vertical deflection plates, namely: (1) a high gain, capacitively coupled amplifier, (2) a medium gain, directly coupled amplifier and (3) direct connections to the deflecting plates. A built-in voltage calibrator for calibrating the sensitivity of the vertical amplifier circuits is actuated by a switch that connects the calibrator to the inputs of the Y-axis amplifier.

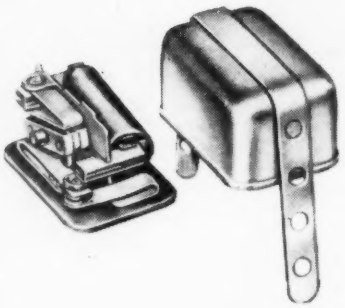
Signals are applied to the horizontal deflection plates through a similar choice of channels, and a fourth position of the horizontal selector switch connects the sweep-circuit output to the amplifier input. Sweep generator produces a wide range of sweep speeds for recurrent and driven operation. These circuits insure a high degree of stability, linearity and synchronizing ease. On driven sweep, automatic beam-blanking circuits intensify the spot during only the go-time. Thereby, photographs of high-speed transients on driven sweep can be safely made with the shutter of the camera open both before and after the occurrence of the transient. The technique of making fog-free film recordings on every attempt is thus greatly simplified. Provision is also made for Z-axis modulation.

The recurrent range of the linear time base is 1 cps to 150 kilocycles per sec, and the duration of the driven sweep is continuously variable from 1 sec to 20 microseconds. Sinusoidal frequency response, identical for vertical and horizontal axes, is: a-c amplifiers—uniform within ±10 percent from 5 cps to 200 kilocycles per sec; d-c amplifiers—uniform within ±10 percent to 200 kilocycles per second.

Thin Electrical Insulation

Johns-Manville Corp., 22 E. 40th St., New York 16, N. Y.

Quinterra, a tissue-thin asbestos-base inorganic electrical insulation, permits appreciable saving of space in electrical equipment. Available in tape rolls from ¼ in. up to 36 in. wide, material varies from 1.5 to 20 mils in thickness. At room temperatures the dielective strength is about 250 v per mil, while at 300 C, this value increases to nearly 400 v per mil. Heat dissipation is rapid from this incombustible insulation that does not char when directly exposed to flame.



Motor Starting Switch

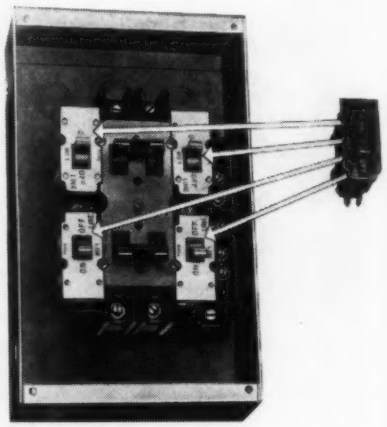
Fitch Allen & Co., 1131 Bryn Mawr Ave., Chicago 40, Ill.

Universal type starting switch can be used with single phase, split phase and capacitor start 90 to 130 v, 50/60 cycle motors in the ⅙ to ⅓ hp range. Unit is connected in series with the working winding and opens the starter winding circuit between 1,400 and 1,500 rpm as the current drops. An external resistance can be used if special circuit opening speeds are desired. Long cover-strap provides flexibility for either internal or external mounting. Overall dimensions are 1⅛ in. wide by 1 1/16 in. long by 1 in. high.

Three-Way Air Meter

Anemostat Corporation of America, 10 E. 39th St., New York 16, N. Y.

Anemotherm three-way meter measures air velocity, temperature and static pressure throughout wide ranges. Velocity range is from 10 to 5,000 fpm and temperature range is from 30 to 155 F. Static pressure is read directly in inches of water from 0.05 to 10 positive and 0.05 to 4 negative. Battery operated instrument with flexible cable and probe weighs 11 pounds. The wide ranges of sensitivity permit application to a variety of fields including heating, ventilating and air conditioning.



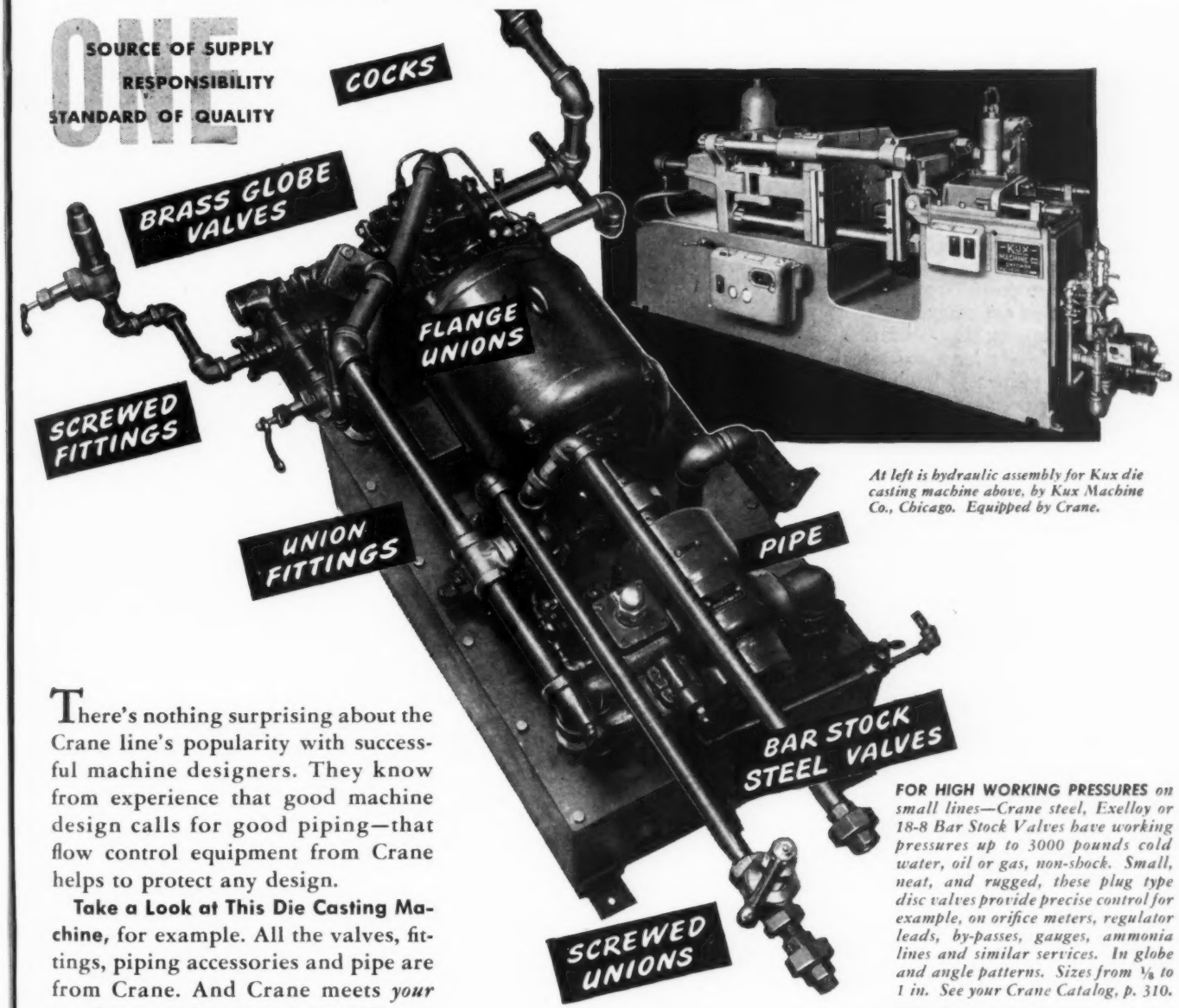
Electrical Load Center

Square D Co., 6060 Rivard St., Detroit 11, Mich.

Type MB4 Multi-breaker service and load center contains high speed, thermal-magnetic (coilless) tripping devices and has sufficient space for instal-

"Designers' Choice" in piping equipment ...the complete CRANE line

ONE
SOURCE OF SUPPLY
RESPONSIBILITY
STANDARD OF QUALITY



At left is hydraulic assembly for Kux die casting machine above, by Kux Machine Co., Chicago. Equipped by Crane.

There's nothing surprising about the Crane line's popularity with successful machine designers. They know from experience that good machine design calls for good piping—that flow control equipment from Crane helps to protect any design.

Take a Look at This Die Casting Machine, for example. All the valves, fittings, piping accessories and pipe are from Crane. And Crane meets *your* requirements just as satisfactorily, regardless of the design on your boards—offers you the world's widest choice of piping equipment ... in brass, iron, steel, and corrosion-resistant alloys.

Relying on Crane as the **Sole Source** for everything in piping simplifies buying and storekeeping, as it does specifying. You don't have to shop around. **One Responsibility** for quality of equipment helps to speed product assembly operations in your plant. And to your customers a Crane-equipped product has added value. For the **High Quality** of Crane equipment stands unsurpassed for more than 90 years.

FOR HIGH WORKING PRESSURES *on small lines—Crane steel, Exelloy or 18-8 Bar Stock Valves have working pressures up to 3000 pounds cold water, oil or gas, non-shock. Small, neat, and rugged, these plug type disc valves provide precise control for example, on orifice meters, regulator leads, by-passes, gauges, ammonia lines and similar services. In globe and angle patterns. Sizes from ⅛ to 1 in. See your Crane Catalog, p. 310.*

Crane Co., 836 S. Michigan Ave., Chicago 5, Ill.
Branches and Wholesalers Serving All Industrial Areas

EVERYTHING FROM ...

VALVES • FITTINGS
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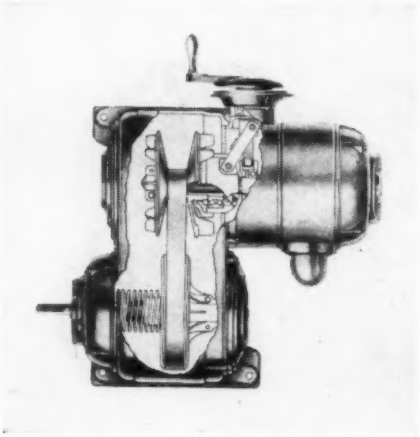
lation of additional circuit components. Assembly consists of a double polarity, four-pole basic unit with provisions to receive four more single pole "Ad-on" units that are plugged in on main terminals. This method of adding circuits to the basic unit reduces installation cost of future circuits by eliminating the wiring, nipples, locknuts and bushings required for installation of additional separate devices. Multibreaker can be used for direct switching of lighting and appliance circuits as well as to provide circuit protection. Single poles are calibrated at 15, 20 and 30 amperes. Double-pole 40 and 50 amp circuits have individual trip. Thermal elements holds harmless, momentary overloads but trips if overload continues.

Magnetic Starters

Ward Leonard Electric Co., Mount Vernon, N. Y.

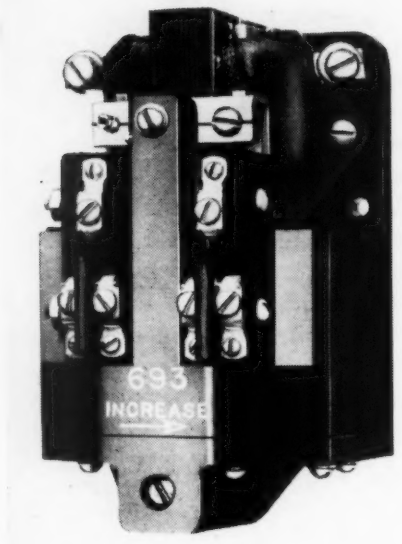
Bulletin 4111 Size 1 a-c magnetic starters are available for non-reversing, across-the-line starting of polyphase squirrel cage induction motors and single phase motors. Units are designed in accordance with Underwriters' and NEMA standards for industrial control, and are suitable for use with separate pilot devices for 2- or 3-wire remote control, or with local control pushbutton stations or selector switches. Starters have a maximum enclosed rating of 7½ hp, 440-550 v, three phase, 60 cycles. Standard operating coils are for 110, 208-220, 440 and 550 v; 25, 50 or 60 cycle service. Dual voltage coils can be supplied for 110/220 or 220/440 v, 60 cycle installations.



Variable-Speed Motors

U. S. Electrical Motors, Inc., Los Angeles 543, Calif.

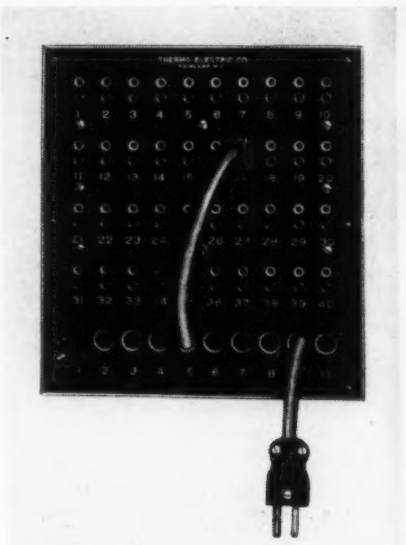
A complete line of compact variable-speed motors with ratings from ¼ to 50 hp have output shaft speeds ranging from 1 to 10,000 rpm. Speed adjustment is made manually, as shown, or by an electrical remote control unit. Spring assembly on lower shaft automatically maintains optimum belt tension. Belt has synthetic fiber cords and sheaves slide on 16-tooth involute splines. Single or double reduction gears can be included to give a speed rating range up to 7 to 1. Horizontal or upright housings are available to fit a variety of installations.



Pneumatic Timer

Cutler-Hammer, Inc., 296 N. 12th St., Milwaukee 1, Wis.

Small a-c pneumatic timer is designed for a wide variety of industrial control timing needs such as machine tools, conveyors and automatic processing equipment where space is at a premium. Timer is available with either one or two independently adjustable timing steps—the two steps eliminate one timing relay. Timing adjustment is easily made from the front of the unit with a screw driver. Eight turns of the screw cover entire timing range from 0.3 sec to 3 minutes. Approximate dimensions are 2¾ in. square by 4⅞ in. high.



Thermocouple Panels

Thermo Electric Co., Fair Lawn, N. J.

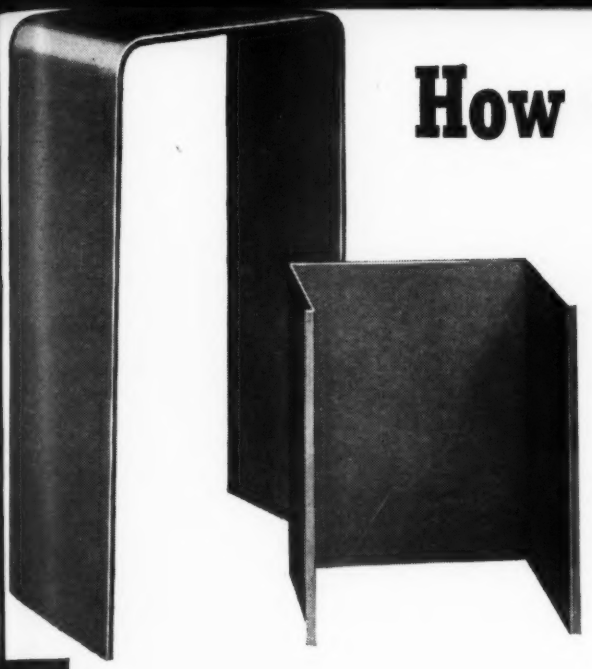
Connector panel provides a quick and flexible method for easily connecting any one of a series of thermocouples to any position on multiple recorders or indicating and controlling pyrometers. Designed for flush mountings, panels are available in various combinations with any desired number of polarized jacks and plugs made of thermocouple materials for all standard calibrations. Position numbers and polarity are indicated on back of panel. Rubber covered cable with a finger-grip, bakelite plug reaches any connector jack on panel. Bakelite plugs have soldered connections and screw-fastened covers permitting replacement of cable.

Plastic Insulating Tubing

Extruders Inc., 8509 Higuera St., Culver City, Calif.

A vinyl-base plastic tubing has been developed for use as a protective insulation in electrical conduit and harness installations. Known as Insulite, tubing has specific applications in the automotive, marine, power and proc-

How to *Form* metal cabinets on a PRESS BRAKE!



By extending bed and ram beyond the ends of the press and installing tangent bending mechanisms, BATH has added cabinet forming to the already wide range of functions of its STURDYBENDER Press Brake.

Now it is possible to do a complete job on one machine—blanking, perforating, notching, bending of flanges *and forming*. Obviously, this method provides a tremendous savings in the handling of materials, and eliminates the need for special forming equipment. The tangent bending mechanism is adjustable, to permit forming a wide variety of shapes. Cabinets for refrigerators, washing machines, deep freeze units, vending machines and other similar products are already being formed by this means.

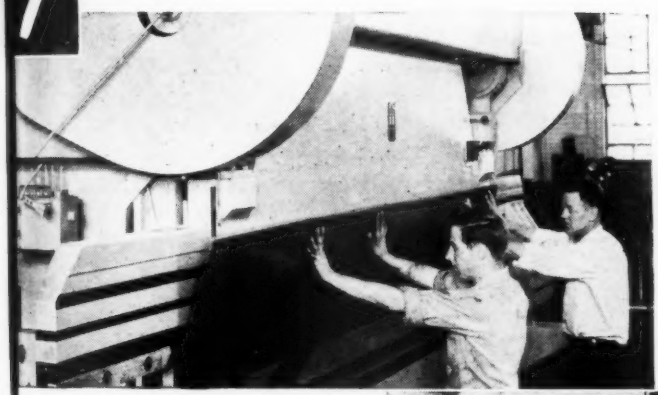
Since more and more manufacturers are finding the STURDYBENDER tangent bending device an economical answer to their cabinet forming problems, you, too, may be interested in the details. A letter will bring full information.

The CYRIL BATH COMPANY

6987 MACHINERY AVENUE

CLEVELAND 3, OHIO

1



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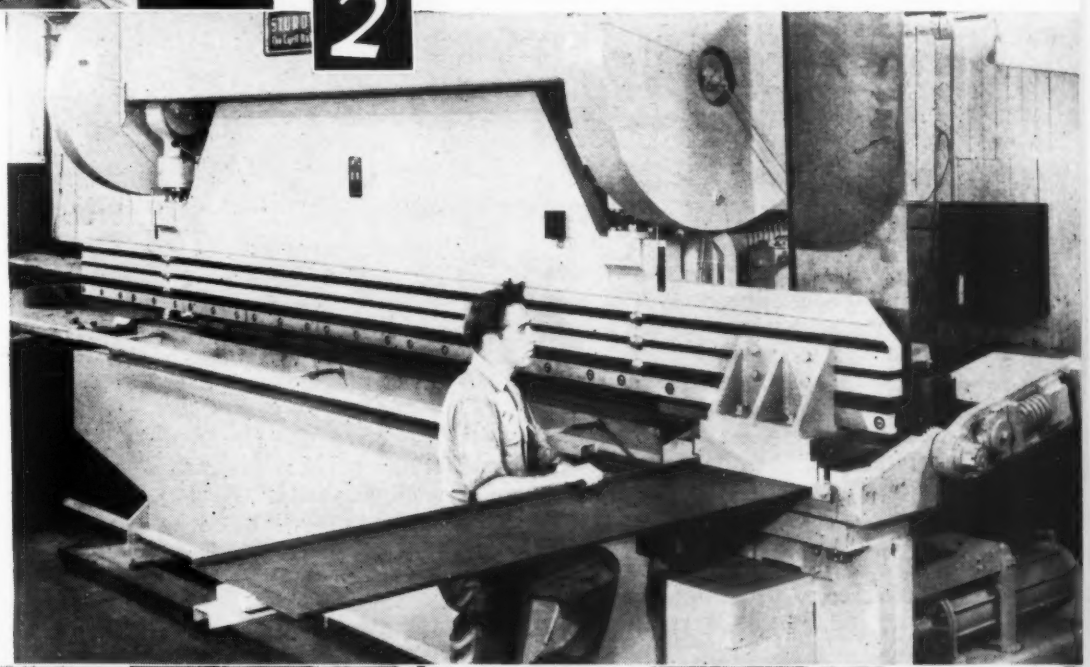


First step in production of refrigerator shell is form- 90° flange and lock flange sides of flat sheet, conven- al press brake operations.

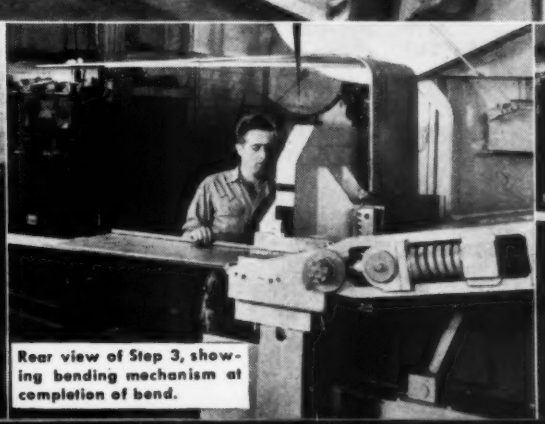
•

et is then inserted in gent bending device the first top corner bend made. Buckling or wrin- g of flanges at the corner ius is avoided.

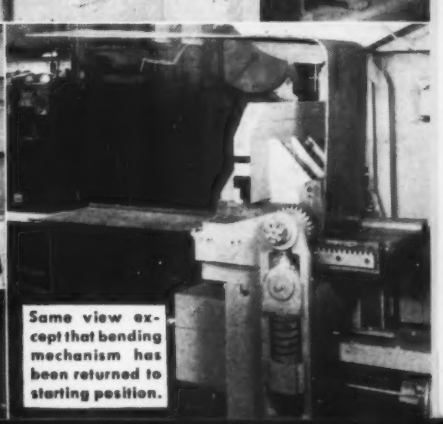
•

et is moved further into machine and second cor- is formed. Once machine et up, duplication of parts is automatic and very accurate.

3



Rear view of Step 3, showing bending mechanism at completion of bend.



Same view except that bending mechanism has been returned to starting position.

essing industries, as well as for transmission of corrosive fluids and gases in chemical equipment. Material is nonflammable, flexible at temperatures down to —65 F, and stable at temperatures up to 260 F; dielectric strength is 800 v per mil. In addition to the natural color, tubing is available in black, white, blue, green, red and yellow. It is also made in tape form in widths from ½ to 1 in. and in thickness from 0.008 to 0.020 inch.

Subminiature Tube

Raytheon Manufacturing Co., Newton, Mass.

Subminiature Type CK 5703/CK 608-CX is a heater cathode triode with an amplification factor of 25 and a rated mutual conductance of 5,000 micromohos. It is suitable for a wide variety of applications including guided missiles and gives good high frequency performance. Plate dissipation is 3 watts and capable output is nearly 1 watt at 500 megacycles. Tube can be soldered into circuit or plugged into commercially available sockets. The flat-press construction minimizes glass defects and damage from mechanical stresses.

Permanent Valve Packing

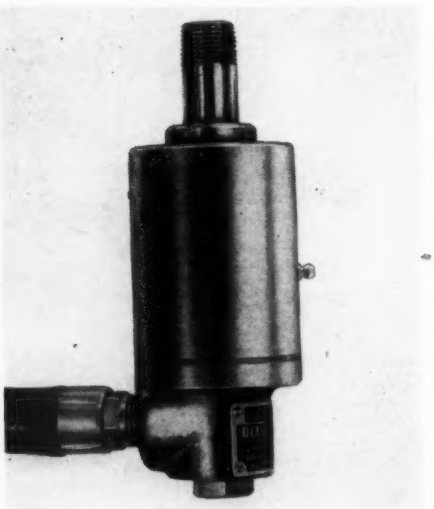
United States Gasket Company, 628 N. 10th St., Camden, N. J.

An easily deformable valve packing, Chemiseal 510, is fabricated from 100 percent unorientated polyletrafluoroethylene Teflon. Packing is positioned in valve stuffing boxes in the form of a sleeve or bushing and, with low nut pressure, flows to the dimensions of the stuffing box. Being chemically inert, material does not require replacement and its anti-adhesive characteristics prevent freezing and binding.

Resilient Ribbed Sheeting

Korfund Co., Inc., 48-15 32nd Place, Long Island City 1, N. Y.

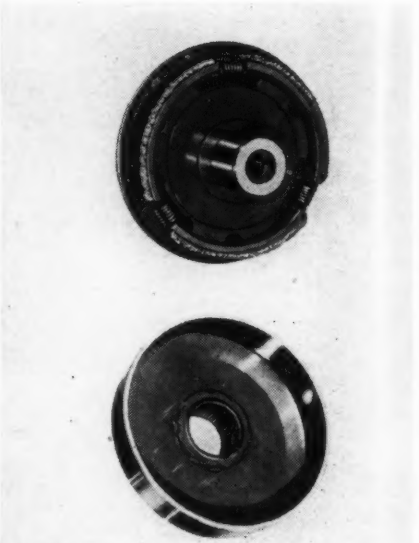
Elasto-Rib sandwich-type sheeting consists of oil-resistant Buna-S synthetic rubber bonded to top and bottom surfaces of high-grade cork. Grooves molded in the rubber are at right angles to reduce vibration and deflection. Average loading for most applications is 20 psi or 2,880 lb per sq ft; with severe vibration, a double layer with ribs crossed is recommended. Sheets 24 x 36 x 1 in. can be cut with knife or circular saw or ordered cut to specified sizes.



Rotating Joint

Deublin Co., Northbrook, Ill.

Two types of revolving joint used for introducing heating and cooling media into rotating shafts, cylinders and drums can handle steam, air and liquids in the temperature range from —30 to +300 F. Double-row ball bearings and special Deub-Lock pressure seal insure smooth, leak-proof operation. The Monoflow type joint permits feeding of the circulated medium into one side of the cylinder or shaft; the Duoflow type both introduces and withdraws it from the same side of the cylinder or shaft. The two types are interchangeable without removal from the equipment on which it is installed. Joints are available in ½, ¾, 1 and 1½ in. standard pipe sizes.



Centrifugal Clutch

Saginaw Products Corp., 57B River St., Saginaw, Mich.

Two sizes of automatic centrifugal clutch are available for use with either electric motors or internal combustion engines. Reversible units are applicable to small powered assemblies such as scooters and loading conveyors using up to 3½ hp drives. High starting torque and relatively low operating torque characteristics are obtainable with engagement speeds ranging from 750 to 2,200 rpm. Smaller clutch has two shoes and a 4¼ in. dia drum casing; larger unit has three shoes and a 5¼ in. casing. Plain or adjustable pulleys can be supplied.

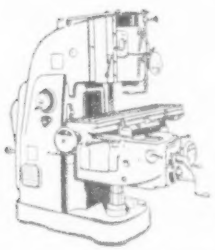
Strain Gage Indicator

Statham Laboratories, 9328 Santa Monica Blvd., Beverly Hills, Calif.

The Model RD battery-operated indicator is specifically designed for measurement of strain in conjunction with standard resistance wire strain gages. Six sets of binding posts and a gage selector permit rapid successive readings of six strain gages attached to various parts of a complex structure. Each set of binding posts consists of one pair for the "live" gage and a second pair for a "dummy" gage that serves to provide compensation for temperature variations. Adjacent to each set of binding posts is a separate balance control for initial balance of the "live" and "dummy" gages. Two ranges are provided: 1000-0-1000 and 10,000-0-10,000 microinches per inch. Accuracy in each case is 0.1 percent of full range. Instrument measures 6⅜ x 11 x 21 in. and weighs 25 pounds.

Extra S.A.*

without extra cost



With the buyer in the driver's seat today . . . purchasing what he *wants* instead of what he can get . . . it's more important than ever that your product have an extra sales appeal—a good-looking finish.

You can meet the demands of this competitive market by using a finish developed for your specific needs by the Advance Paint Company. Advance finishes combine sales-increasing "eye-appeal" with a protective, long-wearing surface.

Your product will take on a better appearance without running up finishing costs, for the Advance finish you use is formulated specifically for your product and for your spray room conditions.

Get the best finish for your metal product at the lowest cost by turning your finishing problems over to the Advance Paint Company for recommendations and formulations.

***SALES APPEAL**—the qualities of a salable product, especially its appearance, performance, and durability, which cause it to be selected by the buyer.

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Ward Leonard Ribflex Resistors Are Made For Fast Heat Dissipation

Size for size, Ward Leonard Ribflex Resistors have 85% to 95% greater wattage rating than ordinary wire-wound resistors. Flat reflexed form of resistance element provides a greater area for heat dissipation. Excellent for both continous and intermittent duty.

Ward Leonard resistors are "Result-Engineered". By modifying a basic design Ward Leonard can often give you the results of a special . . . for the cost of a standard.

Write for Resistor Catalog. Ward Leonard Electric Co., 63 South St., Mount Vernon, N. Y. Offices in principal cities of U. S. and Canada.

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NEW BOOKS

Copper as an Alloying Element In Steel and Cast Iron

C. H. LORIG, *Assistant Director, and* R. H. ADAMS, *Research Metallurgist; Battelle Memorial Institute.* 213 *pages,* 6 *x* 9 *in. Published by McGraw-Hill Book Co., Inc.,* 330 *W.* 42*nd St., New York* 18, *N. Y.* $3.

A reference handbook for the metallurgist, design engineer and others interested in the latest technical information on the use of copper as an alloying agent in iron and steel. The book is a complete summary and critical appraisal of the world's research on the alloys of iron and steel.

Practical data on the properties, characteristics and applications of cast copper steels, wrought copper steels, copper cast iron and copper malleable iron are included.

For the steel man there is a thorough discussion of the manufacture, properties, and uses of steels and cast irons containing copper as an alloy including the important copper-bearing corrosion-resistant materials.

For the researcher there is presented a correlation of all research on the binary and ternary systems containing iron and copper.

For the engineer, the book contains a complete section on the corrosion resistance of low-copper steel, with a review of recent developments in precipitation hardening of copper steels.

Wear and Lubrication Of Piston Rings and Cylinders

REEMT POPPINGA. *Translated by* EDMUND KURZ. 201 *pages,* $5\frac{1}{4}$ *x* $8\frac{1}{4}$ *in. Published by The American Society of Lubrication Engineers,* 343 *South Dearborn St., Chicago* 4, *Ill. Price to members of A.S.L.E.,* $3.; *to nonmembers,* $3.50.

This volume was originally published in 1942 by the Society of German Engineers, Inc. (V.D.I.) It is now published through arrangement with the Office of Alien Property, Department of Justice, Washington, D. C.

The purpose of the textile is to explain and investigate the various difficulties attendant to the causes of wear, and to present a comprehensive picture of the state of research in this field.

Included in the major topics discussed are: Effect of material structures on wear; influence of the lubricant upon wear; influence of engine

operating conditions upon lubrication and the wear of cylinder and piston rings; and general considerations concerning wear.

More than 600 books and periodicals are listed in the appendix as sources for further information pertinent to the subject of wear and lubrication of industrial machinery.

Solid Luminescent Materials

Edited by GORTON R. FONDA *and* FREDERICK SEITZ, *Division of Electron Optics, American Physical Society.* 459 *pages,* 5½ *x* 8½ *in. Published by John Wiley & Sons, Inc.,* 440 *Fourth Ave., New York* 16, *N. Y.* $5.

This book is the result of a cooperative effort on the part of many scientists who are working in the field of luminescence. The papers that appear in this monograph are the outcome of a three-day conference held at Cornell University in October, 1946. The purpose of the conference was to correlate the knowledge acquired under the duress of wartime conditions. The conference provided for a general survey of various phases of luminescence, including the theory of various types of phenomena connected with this subject, as well as the purification, crystalization, and general constitution of luminescent materials.

The eight introductory papers furnish a rapid survey of the field. The topics include, besides the phenomena of fluorescence itself, those phases of phosphorescence that deal with the storage and release of luminous energy. The most novel development in this category is represented by the infrared-sensitive phosphors. The next group of seven papers deals with factors affecting fluorescence characteristics. A third section discusses storage of luminescence energy.

Technical Descriptive Geometry

B. LEIGHTON WELLMAN, *Associate Professor of Mechanical Engineering, Worcester Polytechnic Institute.* 508 *pages,* 6 *x* 9 *in. Published by McGraw-Hill Book Co., Inc.,* 330 *W.* 42*nd St., New York* 18, *N. Y.* $4.

Complete up-to-date treatment of descriptive geometry for students and industrial draftsmen. The text begins with the elementary concepts and progresses to the complex problems of intersection and development of surfaces.

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phasized, especially with respect to surfaces, their representation, intersections, and development. Double-curved surfaces such as are encountered in ship, automobile, and aircraft construction, are given more than the usual space.

One chapter, 143 pages, is devoted to problems associated with subjects developed throughout the text.

In the appendix the author includes a number of plane constructions that have been found to be useful accessories in the solution of descriptive geometry problems.

Marketing by Manufacturers

Edited by CHARLES F. PHILLIPS, *Ph.D. First Edition, second printing. 639 pages, 6 x 9 in. Published by Richard D. Irwin, Inc., Chicago, Ill.* $6.

The complex subject of consumer and industrial products distribution is beyond the field of interest of most engineers. But to those who wish to understand the intangible factors that relate their design efforts to the ultimate sale of a product, this book is a good reference.

This volume is the product of ten authorities specifically for manufacturers. Hence most of the book relates to markets, channels of distribution, market research, advertising and the organization of sales forces. Sections discussing the effect of customer requirements on product design will be of particular interest to the engineer.

Industrial Electronics Reference Book

Prepared by Engineers of the Westinghouse Electric Corporation. First Edition. 680 pages, 8½ x 11½ in. Published by John Wiley & Sons, Inc., 440 Fourth Avenue, New York 16, N. Y. $7.50.

This reference, which is a fundamental handbook that will be valuable both for the electronic specialist and the ordinary development engineer who may be concerned only with obtaining basic information on characteristics of electronic circuits or components, has been several years in preparation. Each of the 36 chapters has been prepared by one or more specialists. The chapter on radio-frequency heating was written by D. Venable and T. P. Kinn, both of whom played a major part in the design of high frequency oscillators for induction and dielectric heating. The chapter on oscillator circuits is the work of Dr. William Altar of the Westinghouse Research Laboratory and Dr. F. U. Condon, formerly Associate Director of the Westinghouse Research Labora-

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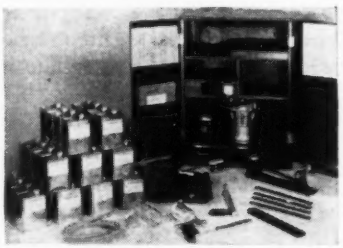

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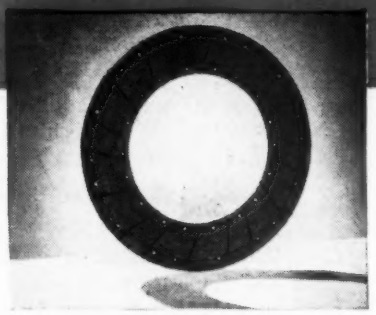

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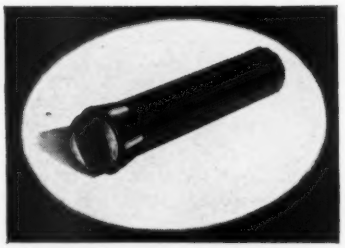

2 **Tapered gasket ring, exposed to acids, alkalies and essential oils, must impart no taste or odor to solutions. Must be free from "cold flow" for tight seal through a temperature range from 0° to 100° F. What material would *you* pick?**

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tory and now Director of the U. S. Bureau of Standards. Other chapters have been written by equally well known engineers.

The first section of the book is devoted mainly to basic electronic theory. It proceeds from the kinetic theory of gases to the control of free electrons and the theory of vacuum and gaseous tubes. The remaining section is broken down into chapters on basic electronic devices such as photo-electric devices, transformers, rectifier circuits, amplifier circuits, electronic instruments, electronic motor controls, inverters, and others. Throughout the text, fundamentals are stressed rather than specific equipment, thus assuring that the text will not be made obsolete by new developments.

An idea of the scope of the book can be had by the break down of the chapter on amplifying circuits. This chapter is devoted to a discussion of transformer-coupled amplifiers, the types of amplifier classifications, a study of voltage and current relationships, anode characteristics of vacuum tubes, and frequency modulation. As are all other chapters, this one is well illustrated with line drawings of circuits, curves of tube characteristics, and photographs of specific applications.

An extensive list of references is given at the end of each chapter to aid the reader in locating additional material on all of the subjects discussed.

Basic Hydraulics

Compiled and edited by: WALTER DORWIN TEAGUE *and Engineering Staff.* 314 *pages,* 6 x 9 *in. U. S. Government Printing Office, Superintendent of Documents, Washington* 25, *D. C.* $1.50.

This basic text on hydraulics was prepared for use in Naval gunner's mate schools. It covers hydraulic principles and tells how they are utilized in various types of hydraulic equipment. Although the book is fundamental and confines itself to non-technical descriptions and simple diagrams, it is an excellent reference for anyone wanting an introduction to the use of hydraulics for machine or equipment actuation.

Particularly interesting are the sections that describe specific components of hydraulic systems. Covered are: gages and meters, pipes, fittings and seals, valves for all purposes, and pumps and motors. The descriptions are written clearly and the diagrams well keyed to the text. Some useful information on hydraulic fluids and systems is given in the last chapters.

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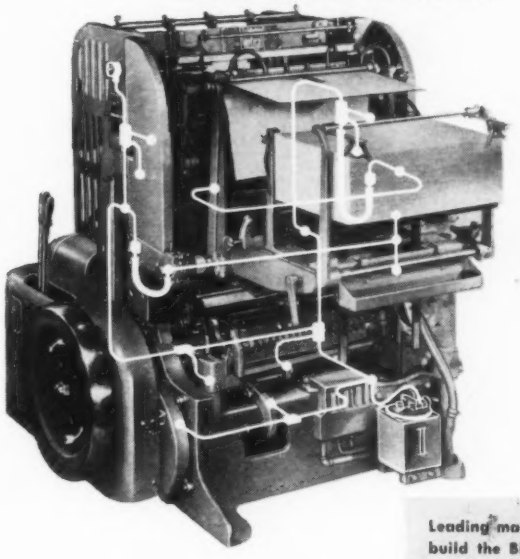

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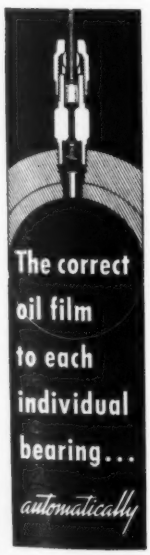

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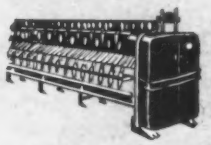

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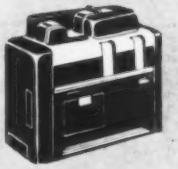

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made to tie in basic hydraulic principles with actual commercial products rather than text book examples.

Fan Engineering

Edited by RICHARD D. MADISON. *Fifth Edition.* 807 *pages* 4½ x 6¾ *in. Published by Buffalo Forge Company, Buffalo, N. Y.* $6.

This handbook combines reference data with enough explanatory text to orient the engineer who is not familiar with fans. Part I covers the properties of air and vapor mixtures, heat transmission and fluid flow. Part II describes the design, characteristics and test methods of various fans. Part III is concerned with the applications of fans in heating, ventilating, air conditioning, air cleaning, combustion draft, and conveying. Part IV goes into the details of fan selection and fan construction. It also describes related equipment such as heaters, coolers and scrubbers.

Design of Machine Elements

M. F. SPOTTS, *Associate Professor of Machine Design, Northwestern University.* 402 *pages,* 6¼ x 9¼ *in. Published by Prentice-Hall, Inc.,* 70 *Fifth Ave., New York* 11, *N. Y.* $6.65.

This two-semester text is written for third and fourth year college students to serve as a source book for engineers in industry. Each chapter is complete in itself, so that they can be studied in any order. Subjects covered include stress concentration, repeated loadings, theory of torsion, springs, screws, bolts, brakes, clutches, welded and riveted connections, lubrication, ball and roller bearings, spur gears, helical, bevel and worm gears, miscellaneous machine elements such as flywheels, gaskets and roller chains. The last two chapters discuss dimensioning, detailing and materials.

Mechanics of Machinery

C. W. HAM, *Professor of Machine Design, University of Illinois, and E. J. Crane, Western Electric Co., Chicago, Ill. Third Edition.* 538 *pages,* 6¼ x 9¼ *in. Published by McGraw-Hill Book Co., Inc.,* 330 *W. 42nd St., New York* 18, *N. Y.* $5.

Vibrations and critical speeds in shafts and properties of the gyroscope are principal additions to the revision of this textbook. Some additional material on cams, gears, flywheels, governors and balancing is also included.

Part I covers mechanism linkwork such as the four-link mechanism, the four-bar mechanism, the slider crank and its variations, the scotch yoke, toggle mechanisms, quick-return mech-

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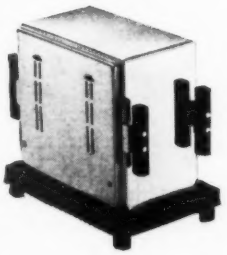

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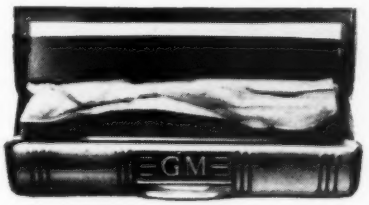

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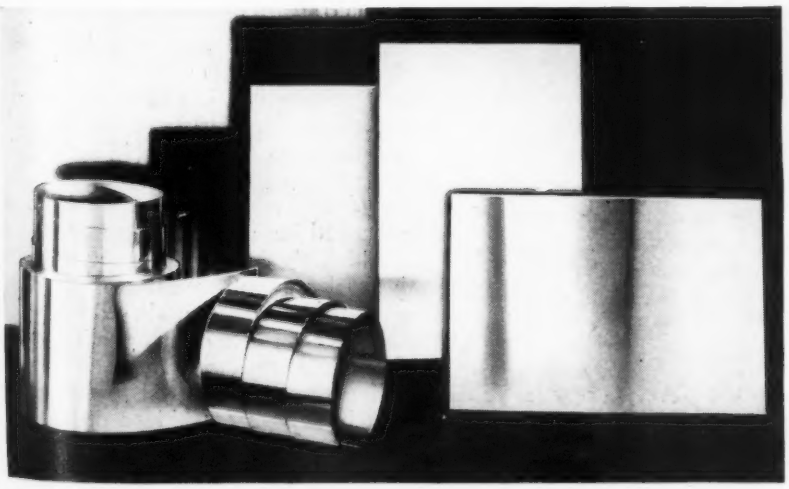

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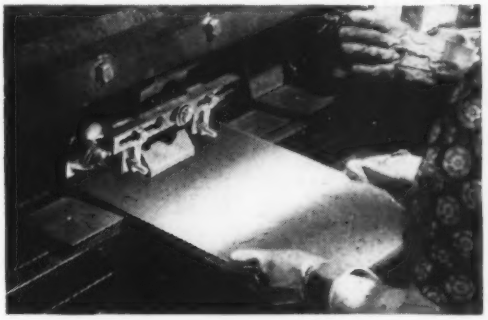

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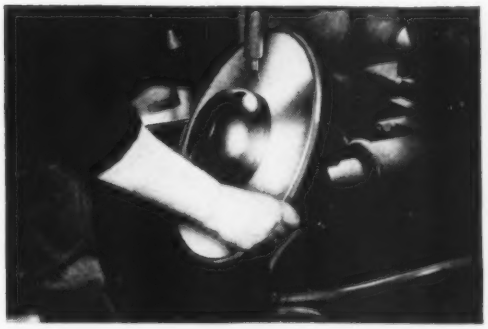

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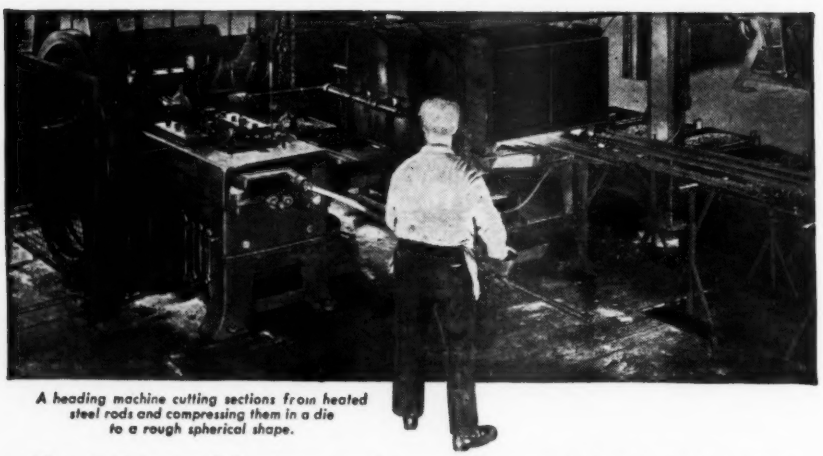

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Kinematics and dynamics of machinery are covered by Part II. Velocities in machines, accelerations in machines, static forces, inertia forces, balancing of machinery, force analysis of a gasoline engine, vibration in shafts and gyroscopic action are analyzed in detail.

Drafting room problems have been selected to show practical application of the theory.

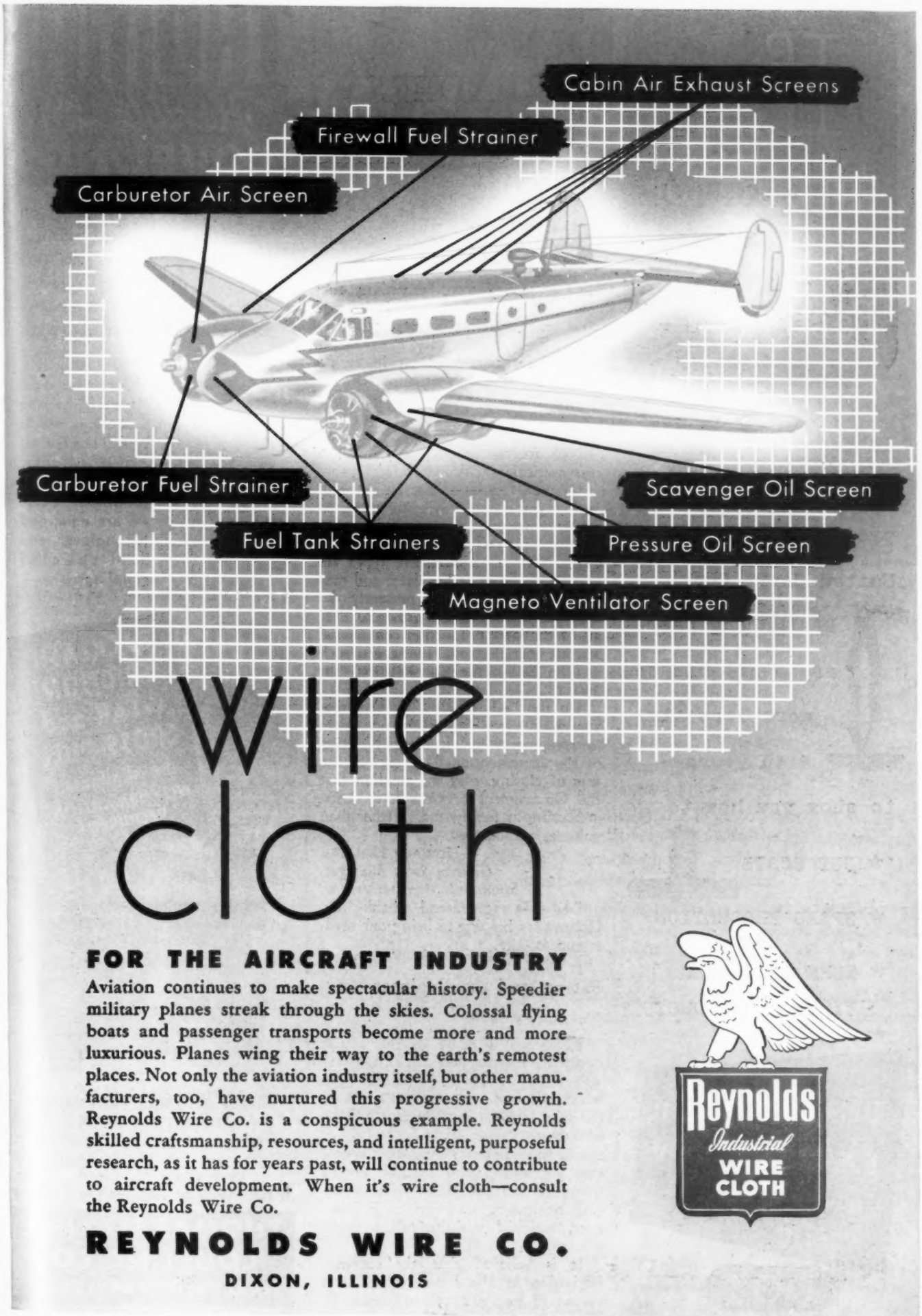
Principles of Jet Propulsion

M. J. ZUCROW, *Professor of Gas Turbines and Jet Propulsion, Purdue University. First Edition. 563 pages, 6 x 9 in. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York 16, N. Y.* $6.50.

This book is based on a series of lectures given by the author in 1943 and 1944 at the University of California. It is an excellent text on the thermodynamics of air and gases and the characteristics of machines for jet propulsion. Although some of the data was assembled from technical literature relating to power plant development for aircraft use, much of the data is applicable for use in other fields where gas turbines and compressors are used.

The author has attempted, with success, the task of assembling both basic and advanced technical data in one book. The first four chapters cover the fundamental characteristics of gases and air, momentum and energy relationships for fluids, thermodynamics of gas flow and the thermodynamic properties of air. The following chapter is an introduction to airplane performance calculations and power requirements; introduced to acquaint the reader with specific power requirements for this type of machine. Chapters 6 through 11 discuss in detail the characteristics of power plant components, including propellers, turbines, compressors, combustion chambers and reaction motors using liquid and solid fuel. The last chapter discusses high temperature alloys, their physical properties and metallurgical aspects.

Although the material contained in the book was gathered from a number of sources, it is logically developed. A uniform nomenclature is used throughout and the illustrations are well prepared and keyed to the text. This book is a valuable contribution to the literature on the thermodynamics of power plants.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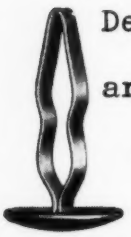

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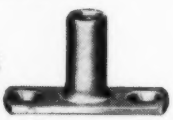

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To the Editor:

In your August 1948 issue of PRODUCT ENGINEERING, we call your attention to a statement regarding fuel gages on page 123 which is part of your article, "Aircraft Instrument Design", by T. B. Holliday.

The aforementioned article, in no small way minimizes the importance of the electronic type capacitor fuel gage. We assume that either the article is of ancient heritage, or the people involved are not properly informed in this particular field.

Of the 31 types of aircraft listed in various publications as being procured under the new 70 Group program, 26 of these aircraft bear electronic fuel gages. The remaining five types are carry-overs from earlier dates and may or may not retain the presently designed-in float gages. Further to this, all of the later production commercial aircraft bear electronic fuel gages, such as, the DC-6, the Constellation, the 240, etc.

On the basis of this, we feel that you must agree facts were not taken into consideration in the publication of the aforementioned article. In the way of closing, may we mention that the Government Services have made measuring of fuel by mass, rather than volume, mandatory.

—HOWARD HOLMES,
General Sales Manager,
Simmonds Accessories, Inc.

Ed.—We agree and thank Mr. Holmes for helping us bring our readers up to date.

Patents and the Engineer

To the Editor:

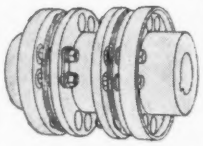
Reference is made to the article entitled "Patents and the Engineer" (Part I) by H A. Toulmin, Jr., which appears on pages 113-128 of the June 1948 issue of PRODUCT ENGINEERING.

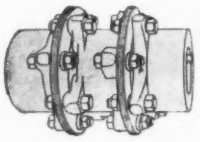
On the whole, I found this article very interesting and helpful for its intended purpose. There is one possibly misleading statement in it, however, which I wish to call to your attention.

It is believed that the sentence beginning at the lower right hand corner of page 123 should read: If

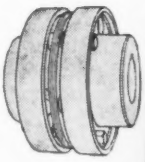
THOMAS Flexible ALL METAL COUPLINGS

Engineered to stand up on the toughest jobs, Thomas Flexible Couplings do not depend on springs, gears, rubber or grids to drive. All power is transmitted by direct pull.





The standard line of Thomas Couplings meets practically all requirements. But if unusual conditions exist we are equipped to engineer and build special couplings.



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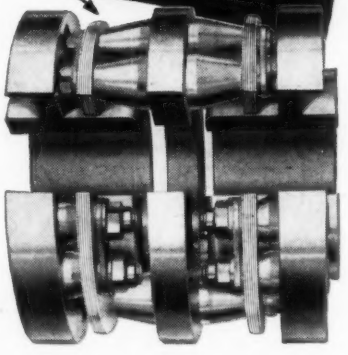
PATENTED FLEXIBLE DISCS



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WARREN, PENNSYLVANIA

A NEW CLARE
26-Point Stepping Switch

For Selection - Sequence Control - Counting - Totalizing

Selection of any channel or circuit path from a total of 26 or 52 circuits is provided by this new CLARE Stepping Switch.

This selection may be at the rate of 30 steps per second on remote control—up to 60 steps per second on self-cycling operations. Operating at these speeds, the switch gives a minimum life of 5 million half-revolutions or 130 million stepping operations.

Each of the ten levels possible for the Type 26 Switch, or the five levels of the Type 52 Switch, is unit-molded in Bakelite. Hand positioning of individual contacts is thus eliminated, and each bank level is easily replaced if a contact becomes damaged in service.

In operation, a pair of double-ended wiper springs is stepped over each bank level of 180 degrees. One end of the wipers is engaged with the bank contact at all times, one end is always free of the bank. The stepping magnet may be remotely controlled or wipers may be stepped automatically by interrupting the magnet circuit through a pair of interrupter springs. As many as eight auxiliary interrupter springs may be provided for other control or signal functions.

Like many other CLARE developments, this new stepping switch was designed to meet a specific requirement . . . has provided an answer to others. Whatever your relay problem, it will pay you to submit it to CLARE. Sales engineers are located in principal cities for your convenience. Look in your classified telephone directory . . . or write to C. P. Clare & Co., 4719 West Sunnyside Ave., Chicago 30, Illinois. In Canada: Canadian Line Materials Ltd., Toronto 13. Cable address: CLARELAY.

STANDARD SPECIFICATIONS

OPERATION:
Automatic (self-interrupting) or remote controlled.

WIPERS:
One to ten, traversing individual contact levels.

INTERRUPTER SPRINGS:
Form 1B (to open the operating circuit at the end of each step). Contacts are single platinum-iridium.

OPERATE SPEED:
Remote controlled operation: maximum 30 steps per second. Self cycling operation: average 60 steps per second, with 48-volt power supply.

FINISH:
Framework and armature: cadmium; Bank contacts and wipers: phosphor bronze.

MOUNTING:
Frame drilled and tapped at each end to accomodate No. 8-32 mounting screw.

DIMENSIONS:
Overall lenght: 6-9/16 in.; width: 2-3/8 In.; height: 4-5/8 in.

NET WEIGHT:
27 oz., approximately.

SHIPPING WEIGHT:
4 lbs., approximately.

Write for Clare Bulletin 101 on complete details.

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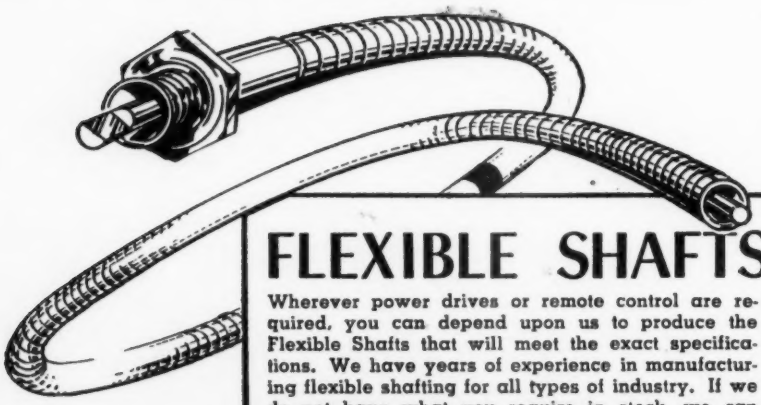

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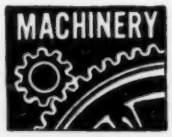

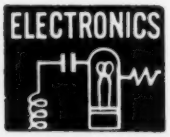

Many new uses for flexible shafts that carry power around any corner have been developed by our engineers . . . in machine shops, electronic, automotive, aircraft, in all industries where power drives or remote control are required. *Write for Manual P. E.*

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the invention has been publicly made, used, or sold in the United States *more than two years* before the patent application was filed, or before the actual date of the invention. . ." The two-year period referrd to in the sentence as printed applies only to patent applications which were filed prior to August 5, 1940, as is clearly brought out by Rule 46 of "Rules of Practice in the United States Patent Office."

If reprints of Parts I and II of this article are to be available, I would appreciate being advised regarding the price of individual as well as of 100 copies.

—ALBERT R. HODGES,
Patent Attorney,
Stromberg-Carlson Co.

Ed.—Through an oversight in correcting the proof, the change to "one year" was not carried over into the final article.

To the Editor:

I would like to take this opportunity to extend my compliments both to you and the author on the article "Patents and the Engineer, Part II". I have already ordered several reprints for distribution in our Engineering Department.

As you suggested in the article, I am writing for additional information relating to the condensed chart on page 127 entitled "Vital Data for Securing Foreign Patents."

—T. R. CLAFFEY, *Patent Dept.,*
The United States Time Corp.

Ed.—A copy of the original complete chart and specific information can be obtained from the author, Mr. H. A. Toulmin, Jr., 308 W. First St., Dayton 2, Ohio.

Plastic Aircraft Wings

To the Editor:

In the Progress in Review section of the June PRODUCT ENGINEERING you mentioned a plastic aircraft wing having a low pressure laminated skin and a honeycomb core.

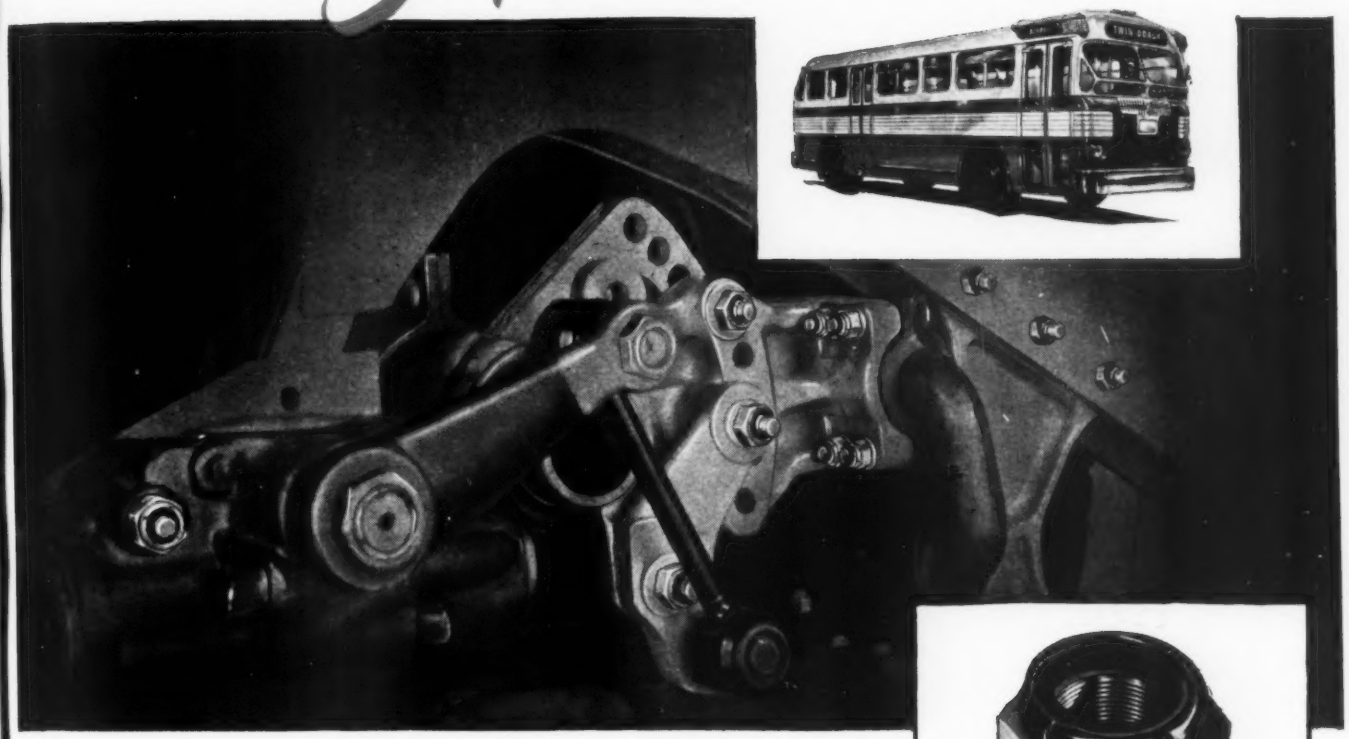
This item was of particular interest as I am writing a thesis for a B.S. degree on the use of plastic faces in sandwich construction for primary aircraft structures.

I would appreciate receiving additional information on this subject.

HOWARD S. GREEN,
Parks College of Aeronautical Technology

Ed.—Consult the March 1948 issue of the British publication *Plastics* for the original article. General engineering data on sandwich laminates can be obtained from the Owens-

TOUGH *Torsilastic Suspension* **HELD PERMANENTLY SECURE**



—*the Red Elastic Collar holds each positioned setting against* VIBRATION! SHOCK!

The toughest detachable fastener connections on the Twin Coach Torsilastic Spring Suspension are on the rubber bonded shackles. These fasteners must be self-locking without any frictional aid from bolt tension or seat pressure. They must maintain positioned settings against the constantly flexing rubber.

Exhaustive tests proved that ESNA Elastic Stop Nuts—with the self-locking, self-sealing Red Elastic Collar—hold tight permanently. They also hold secure the equally important prestressed metal to metal assemblies where accurate bolt loading is essential.

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Here's a challenge: Send us complete details of your toughest bolted trouble spot. We'll supply test nuts—FREE, in experimental quantities. Or, if you want further information, write for literature. Elastic Stop Nut Corporation of America, Union, New Jersey. Representatives and Agents are located in many principal cities.

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Threadless and permanently elastic, it provides these 4 outstanding features:

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3. Provides for accurate BOLT LOADING
4. Seals against LIQUID LEAKAGE along the bolt threads

AND IT IS RE-USABLE

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Corning Fiberglas Corp., Toledo, Ohio and the Air Materiel Command, Wright Field, Dayton, Ohio.

Electrical Grade Laminate

To the Editor:

Please send me further information on the electrical-grade laminate, which is produced by a two-resin treatment as described in the Progress in Review Section of the June PRODUCT ENGINEERING.

I would also like further data on the redesign of Army ordnance equipment as reported by Lt. Gen. R. A. Wheeler, Chief of Engineers of the United States Army and the new alloy that is twice as strong as present alloys and 12 percent lighter.

—C. W. BESSERER,
Supvr. Design Engineering,
The Johns Hopkins University

Ed—The laminate was developed by Professor L. F. Rahm of Princeton University, Princeton, N. J. and additional experimentation is being carried on by Mr. John Alfers, Bureau of Ships, Navy Dept., Washington, D. C.

The redesign of Army Ordnance equipment was discussed by Lt. Gen. R. A. Wheeler, Chief of Engineers, U. S. Army, Washington, D. C. in a talk before the Magnesium Association in March 1948. A copy of this talk can be procured from the Association.

Should Engineers Do Military Research?

To the Editor:

It appears that engineers, as judged by technical publications and meetings, seldom are concerned about whether their work contributes to the welfare of humanity or whether it is morally and ethically justifiable.

What concerns me most is the willingness of engineers to contribute toward the destruction of civilization by working on military weapons without questioning the morality of what they do. It should not require a tender conscience for one to recognize that engineering work on weapons of human slaughter is quite immoral.

At present, such work is justified by pointing with alarm to Russia, but there can be no moral progress if we relate the evils, or what the propagandists say are the evils, of the worst nation on earth and then try to outdo them.

Engineers try to escape personal responsibility by passing the buck to "my government," but the United States has been trying to impress on Germans that they were responsible

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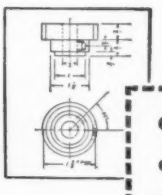
You can see the Model 21 Bruning Whiteprinter and other Bruning equipment at the National Business Show, Booths 219 to 222, Grand Central Palace, New York, October 25 to 30 inclusive.



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for what they did even though it would have meant an absolute refusal to do what German law required. Should not we do what we demand of others?

As for what is to be done about oppressors and aggressors, we should admit that Americans haven't all the know-how and learn from India and Gandhi the civil disobedience techniques and economic weapons used so effectively in their struggle for freedom. —WALTER GORMLY, *Mechanical Engineer, Mount Vernon, Iowa*

Nylon Bearings

To the Editor:

Will you kindly send me information on the nylon bearings described in the "Highlights" section of your July issue? — WILLIAM J. HOGAN, *Chief Engineer, Stapling Machines Co.*

Ed.—The nylon bearings are manufactured by Maclurecraft, Inc., Whitman, Mass. Application information on this material can be gotten from E. I. duPont de Nemours Co., Inc., Wilmington 98, Del.

Plastic Seat Covering

To the Editor:

Please furnish me with further information concerning polyvinylchloride seat webbing, as described on page 104 of the June PRODUCT ENGINEERING.

—R. H. STEVENS, *Development Engineer, Spring Div., The Murray Corporation of America*

Ed.—The Thermo-Artisan Engineering & Mfg. Co., Los Angeles, Calif. can supply full technical information regarding this material.

Glass Fiber Laminates

To the Editor:

Your magazine has been very useful to me in supplying information regarding the uses of bonderized glass or other fibers.

I am primarily concerned with the problem of coloring these materials. Would you be able to give me some detailed information on this phase of the subject?

—W. E. ANDREWS, *Honolulu 11, Hawaii*

Ed.—A wide variety of commercially available colored resins are generally used in these laminates. Detailed information can be obtained from the Owens-Corning Fiberglas Corp., Toledo, Ohio.